DEVELOPMENT OF AN AUTOMATIC TENSIONING CONTROL SYSTEM FOR A REPLENISHMENT-AT-SEA WINCH ARRANGEMENT.

by

George Lewis Shillinger

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

ABSTRACT

The tensioned modified housefall method of replenishment-at-sea provides a system by which a cargo load can be transferred from one ship to another underway at sea through the use of a simple wire rope arrangement. This system requires two winch drums on one of the ships and provides fine control of cargo trolley movement as related to either of the two replenishing ships. A major problem in the realization of an operating replenishment-at-sea system of this type is the development of a satisfactory automatic tensioning control. Wire tension must be maintained within a narrow range during the replenishment-at-sea operation in spite of the individual movements of the two replenishing ships, the transit movement of the cargo trolley from ship to ship, the varying separation between the two ships involved and other disturbances. If wire tension becomes too high the wire will part. Since wire tension is used to suspend the trolley load a loss of wire tension during the replenishment-at-sea operation will cause the trolley load to strike against ship structure or to drop into the sea. The goal of this research effort is the development of an automatic tensioning control for a contemplated operating model of a tensioned modified housefall system.

ACKNOWLEDGEMENT

The author is sincerely grateful to Professor Yasundo Takahashi whose suggestions, guidance and interest have greatly assisted and contributed to this research.

I. INTRODUCTION

Basic requirements for an automatic tensioning control are presented in Appendix A. This appendix starts with the fundamentals for maintaining constant tension in a cable stretched between two points with the distance between these two points varying. From this basis Appendix A presents the development and description of the tensioned modified housefall replenishment-at-sea arrangement which possesses the capabilities described in the abstract. Appendix B, taken from previous material, further illustrates the operation of the tensioned modified housefall arrangement. Appendices C and D, also material prepared previous to this report, describe the model system which this report concerns and the status of the development of this model prior to the research effort reported herein. Relative to the development of controls for the contemplated model Mark II it should be apparent, following review of Appendices C and D, that the next logical step toward realization of this model is the more extensive analysis and design of the automatic tensioning control system. Appendix E contains calculations supporting the results of this report.

II. METHOD OF ATTACK

Initially a general control arrangement was designed along the lines of the automatic tensioning control described in Appendix D. With the exception of the servo valve, the hydraulic components selected are all commercially manufactured items. Elasticity relationships and free body diagrams were studied to determine the effects of changes in the separation between two ships, of changes in the lengths of the wire ropes, and of changes in the trolley load position upon the tension in the rig wires. A transfer function relating tension changes and changes in the lengths of wire rope within the replenishment-at-sea rig was developed. A block diagram of the entire automatic tensioning control system was constructed. The open loop portion of this block diagram was established as the product of two principal separate transfer functions: the wire rope sub-system and the mechanical-hydraulic combined winch and control sub-system. The latter transfer function was approximated with lumped impedances. With adjustment of some of the control parameters a stable transfer function evolved for the open loop system. Thus preliminary indications of the sizes of mechanical and hydraulic control components for a stable automatic tensioning control system were obtained.

III. DESCRIPTION OF CONTROL ARRANGEMENT

Figure 1 is a schematic of the automatic tensioning control system studied. If the tension in the wire rope rig increases above the desired value the torque loading upon the winch drums is increased thereby similarly increasing the hydraulic pressure on the pay-in side of each hydraulic motor. Thus the hydraulic pressures P_a and P_b are increased further advancing the pistons and rods of the differential actuators against a common compression spring. Increased hydraulic pressures $(P_a + P_b)$ cause the piston rods to advance in direction A. Hydraulic servo flow is established to the A end of the control actuator while the B end of the control actuator is drained. The control actuator rod moves thereby positioning the stroke control mechanisms of the variable displacement hydraulic pumps in the proper direction to generate main hydraulic transmission flow q. This flow rotates the rotors of the hydraulic motors to pay out wire from the winch drums. Thus the rig tension is reduced. Movement of the control actuator rod also positions a sleeve of the servo valve which nullifies servo flow to the control actuator as the control actuator rod approaches the position dictated by the servo valve spool position. Tension is thereby restored to the

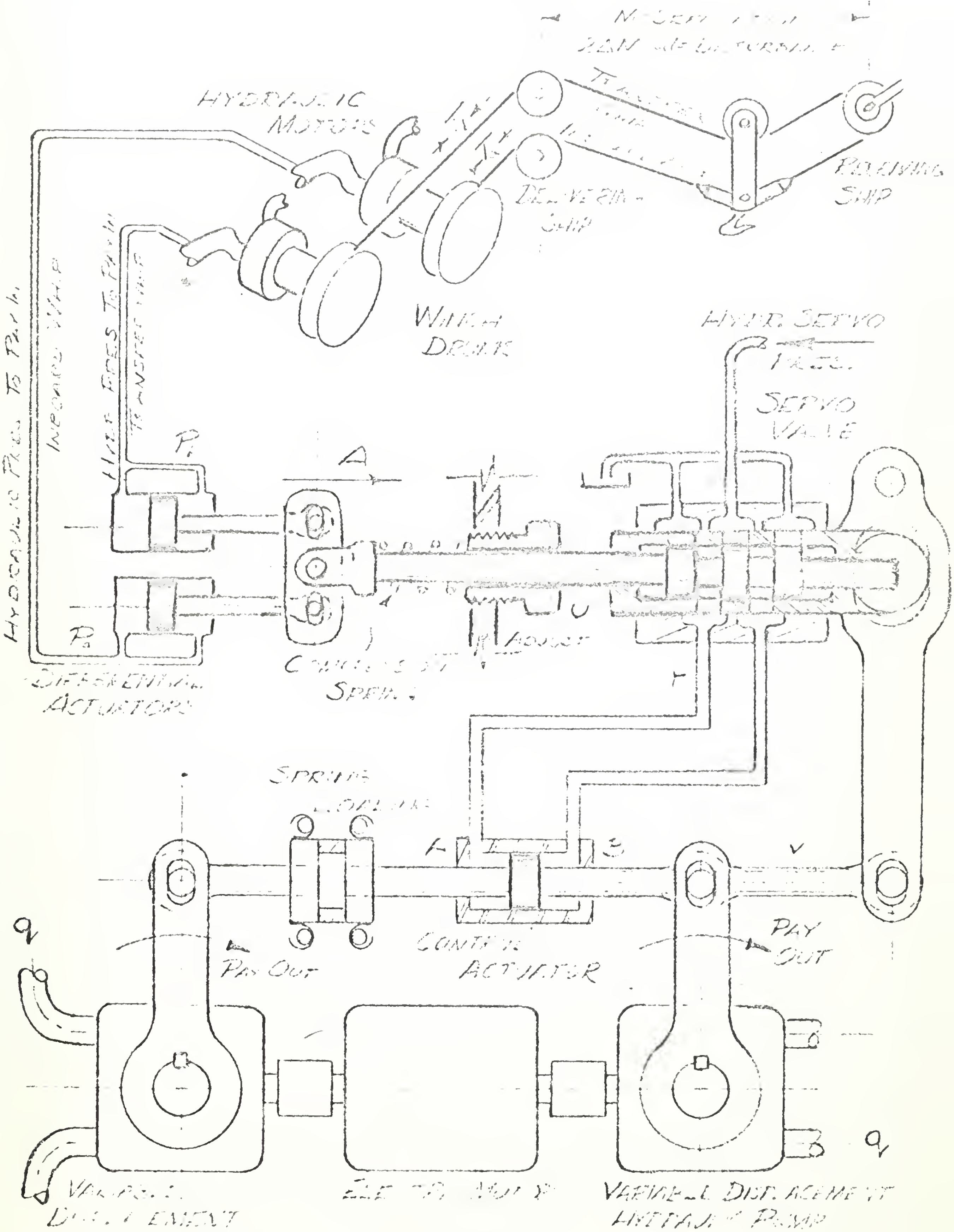


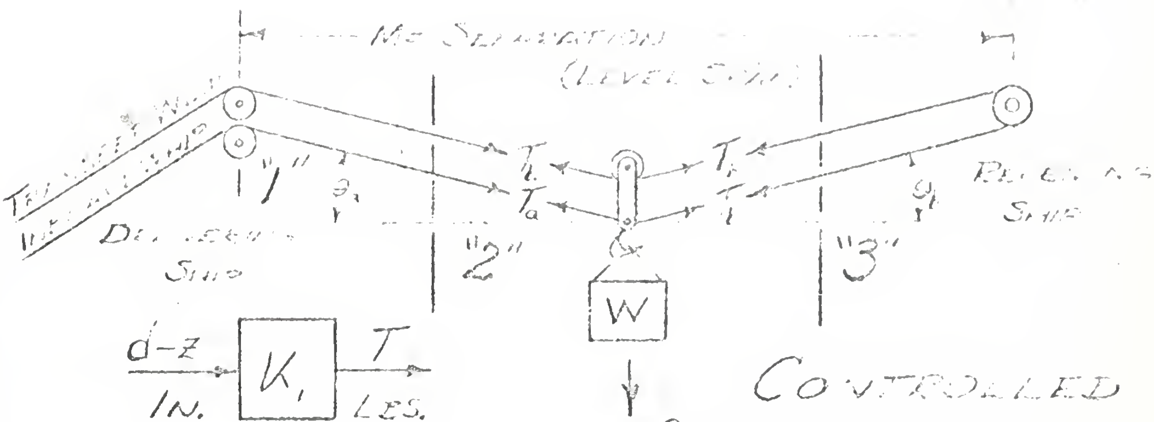
FIGURE 1
AUTOMATIC TENSIONING CONTROL SCHEMATIC

desired value. The automatic tensioning control operates in a similar but opposite manner if rig tension decreases below the desired value.

IV. TENSION-LENGTH RELATIONSHIP

Figure 2 illustrates the effects of varying separation and of cargo trolley position upon the relationship of tension to wire rope length. Also tabulated with figure 2 is a comparison of methods of controlling tension: whether tension should be held constant in the transfer whip, or in the inboard whip, or whether the sum of the tensions of both whips should be maintained constant. The results tabulated with figure 2 were obtained from studies of free body diagrams and elasticity. Note that the ratio of the change in wire rope tension over the corresponding change in wire rope length varies from 14.52 lbs./in. to 90.2 lbs./in., the extremes being the maximum separation (200 ft.) with the cargo load at mid-span and the minimum separation (40 ft.) with the cargo load at either terminal (or without a cargo load) respectively. From the tabulation also with figure 2 it is apparent that the best tension quantity to control is the sum of the tensions of both whips. Limiting maximum tension to 330 lbs. on either of the whips by maintaining a constant sum of tensions on both whips a load carrying capability of 245 lbs. and a tension differential of 37.3 lbs. in each whip during transfer is obtained as compared to corresponding values of about 230 lbs. with a 92 to 95.5 lbs. tension differential in one

WIRE ROPE: $\frac{1}{8}$" 7x19 STAINLESS STEEL CABLE
ULTIMATE BREAKING STRENGTH: 1760 LBS
APPROXIMATE WEIGHT: 0.029 LBS/FT.
MODULUS OF ELASTICITY: 14,000,000 PSI
FOR 5:1 FACTOR OF SAFETY T_{max} = 352 LBS.



	K_1 (LBS./IN.) FOR M = 40'	100'	200'
"1"	90.2	36.0	18.0
"2"	73.0	29.1	14.5
"3"	90.2	36.0	18.0

CONTROLLED CONSTANT (T = 330 LBS)

		T_a = 285 LBS	T_b = 285 LBS	T_a+T_b = 613 LBS
"1"	T_1	288	330	320.5
	T_2	234.5	280.6	292.7
	θ_1	20°41'	21°56'	23°55'
"2"	$T_a = T_b$	236	280.6	306.5
	$\theta_2 = \theta_1$	11°32'	11°32'	11°32'
"3"	T_a	288	236	285
	T_b	330	286.5	330
	θ_1	20°27'	23°35'	21°48'
	ΔT_a	0	92	37.3
	ΔT_b	95.5	0	37.3
	*W	230.7	239.2	245

*W FOR EQUAL CATENARY DEPTH (REGION "2")

FIGURE 2

TENSION-LENGTH RELATIONSHIP

whip if the tension in the other whip is maintained constant. The preceding analysis was based upon static conditions where the tension quantity could ideally be maintained constant. With actual operating conditions the situations analysed would be the upper extreme of the range within which the tension quantity could be maintained under the most severe circumstances for which the system is to be designed. The tension-length transfer function is developed in figure 3. This transfer function is a function of rig geometry and elasticity. Rig geometry, in turn, is a function of cargo weight and cargo position, while elasticity is a function of the total wire rope length involved. Development of this transfer function is based upon the following assumptions:

(a) The wire rope is weightless.

(b) Elastic theory (Hooke's law) is valid. The modulus of elasticity E = 14,000,000 psi and the wire rope area = 0.00616 sq. in.[1]

(c) The wire rope system is not influenced by dynamics.

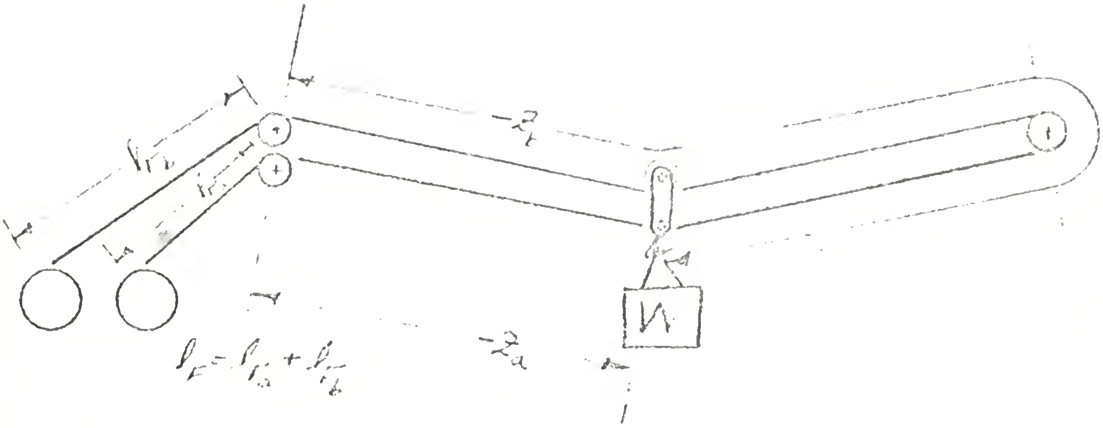
(d) The influence of cargo position varies as the product of the proportional distances of the cargo from each terminal.

(e) The influence of cargo position varies directly as gross cargo ~~load~~ weight.

(f) The terminal locations are at the same height.

$$\xrightarrow[\text{IN.}]{d-z} \boxed{K_1} \xrightarrow[\text{LBS.}]{T}$$

LBS/IN.



$K_1 = f$(CARGO WEIGHT, CARGO POSITION, WIRE ROPE LENGTH)

$$K_1 = \left[1 - \frac{k_G W}{(z_a+z_b)^2}(2 z_a)(z_b - z_a)\right]\left[-\frac{A_{WR}E_{WR}}{z_a+z_b-l_F}\right]$$

WHERE:

k_G = GEOMETRY FACTOR; (0.00314) DERIVED FROM RATIO: $\frac{K_{1(WMS)}}{K_{1(NL)}} = 0.808$ FOR $\frac{W}{T_a+T_b} = 0.4$

. WHERE: $K_{1(WMS)}$ = K_1 CARGO LOAD AT MID-SPAN

$K_{1(NL)}$ = K_1 WITHOUT CARGO LOAD

W = CARGO LOAD WEIGHT, (LBS)

$-z_a$ = LENGTH, INBOARD WHIP, (IN.)

$-z_b$ = LENGTH, TRANSFER WHIP (IN.)

l_F = LENGTH, FAIRLEADS (ASSUME 360 IN.)

A_{WR} = AREA, WIRE ROPE (0.00616 SQ. IN.)

E_{WR} = MODULUS OF ELASTICITY, WIRE ROPE (1.4×10^7 PSI)

$$K_1 = \left[1 - \frac{(0.00628) W z_a (z_b - z_a)}{(z_a+z_b)^2}\right]\left[-\frac{86300}{z_a+z_b-360}\right]$$

FIGURE 3

DEVELOPMENT OF K_1

(g) The wire rope tension sum is maintained constant: ($T_a + T_b = 613$ lbs.).

V. MECHANICAL-HYDRAULIC SUB-SYSTEM

Figure 4(a) is a block diagram of the combination rig and control system. In figure 4(b) and 4(c) the block diagram is adapted for analysis utilizing lumped impedances. In figure 4(d) the transfer function relating the change in "pay-in" pressure (sensor pressure P_s) to the main drive hydraulic flow (F_p) that this change of "pay-in" pressure causes. The transfer function $\mathcal{G}_{\textcircled{1}}$, relating tension changes to corrective wire rope flow (F_W) is developed in Appendix E. This transfer function is based upon the following assumptions:

(a) Parameter values are true.

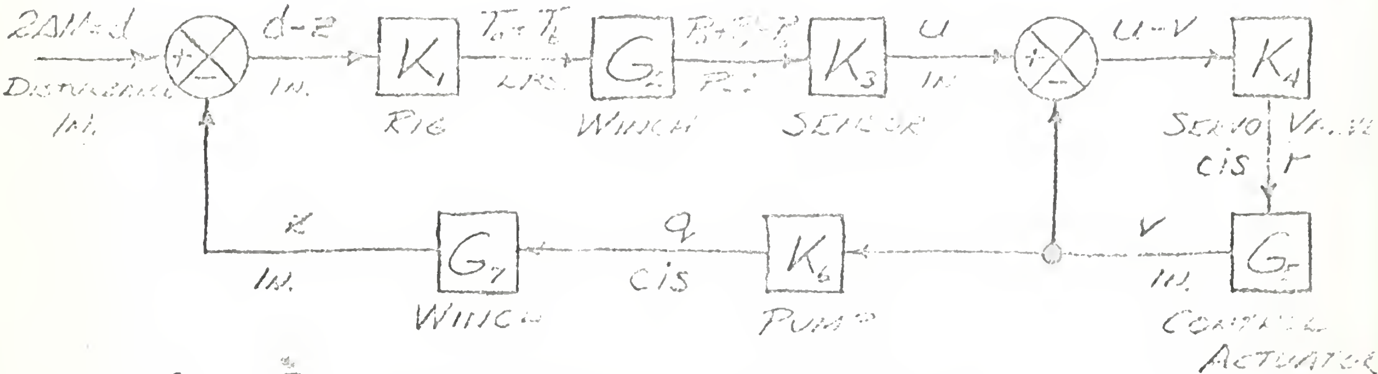
(b) Lumped impedance approximations are effective.

(c) Control linkages have no inertia, friction or stiction.

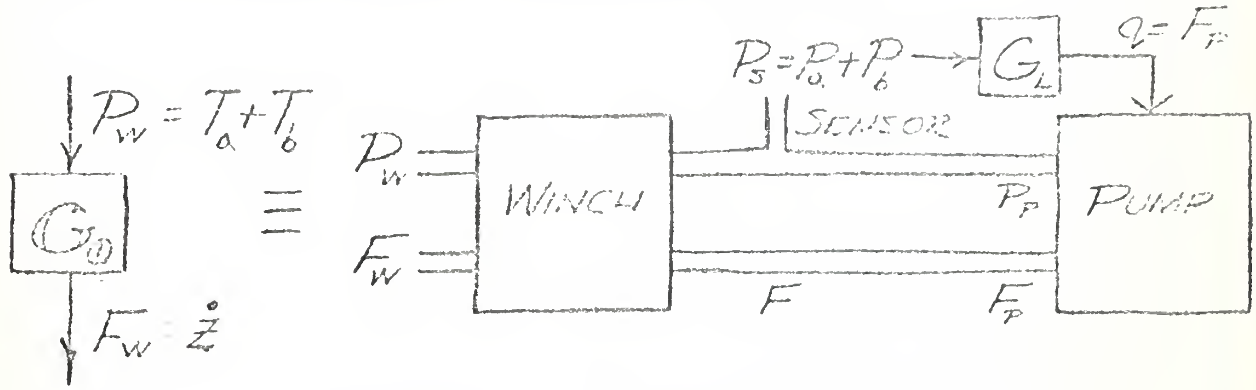
(d) Servo valve has the linear characteristics calculated.

(e) Electric prime mover operates at 1700 RPM.

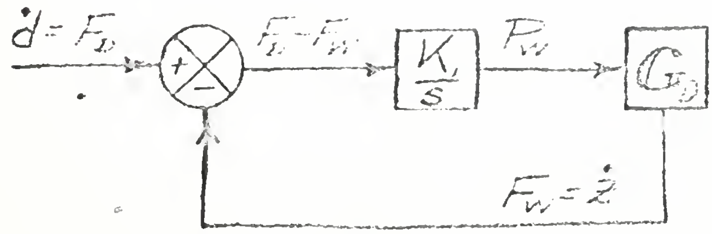
(f) Volumetric efficiencies of hydraulic components are as estimated.



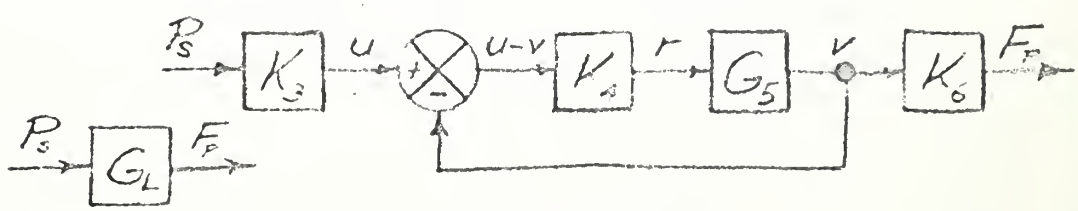
(a) RIG & CONTROL SYSTEM



(b) LUMPED IMPEDANCE: CONTROL SYSTEM



(c) LUMPED IMPEDANCE: RIG & CONTROL SYSTEM



$$G_L = \frac{K_3 K_4 G_5 K_6}{1 + K_4 G_5}$$

(d) SENSOR PRESSURE - MAIN DRIVE HYDR. FLOW

FIGURE 4

BLOCK DIAGRAMS

VI. CONCLUSIONS

Based upon a stable transfer function determined through the use of lumped impedance approximation, the initial design of an automatic tensioning control system was completed including the sizes of hydraulic components, spring rate and variable displacement hydraulic pump stroke control lever lengths.

The transfer function derived in this report should be considered a first approximation relative to the construction of the automatic tensioning control of this report. In assembling and testing a prototype of this model provision should be made for parameter adjustment. With the prototype model it should be possible to readily exchange the compression spring in search of the best suited spring rate to use; and to change the lengths of the control levers which stroke the variable displacement hydraulic pumps. Other parameter adjustments are feasible but with more difficulty. For instance, hydraulic components in the control arrangement could be replaced by larger or smaller sized units should this be necessary.

Future efforts could include:

(a) Review, factoring, simplification, further development and evaluation of the transfer functions.

(b) Analysis of the effects of parameter changes upon the system (such items as changes of hydraulic oil temperature, changes of load inertia, inertia effects of tensioning correction phased to the delivering ship as compared to tensioning correction phased to the receiving ship, inertia effects of cargo transit, effects of correcting tension against the load as contrasted with correcting tension with the load (overhauling load) as related to system stability and the capability of maintaining constant tension, effects of winch drum reversals, etc.).

(c) Computer simulation and analysis with inputs simulating disturbances such as ships' rolls. (Ships' rolls can be conveniently represented by sine waveswith the same periods as the real ships. Thus simulation of the combined rolls of two replenishing ships can be provided by summing two sine waves).

(d) Actual construction of the real model with operating tests, analysis and evaluation.

(e) Study, analysis or construction of other methods of automatic tensioning control.

(f) Analysis with respect to a performance criteria.

(g) Reliability studies and development of fail-safe features.

A suggested performance criteria is the minimization of the difference between the maximum and the minimum values of the tension encountered under actual circumstances in the

most severe environment in which the replenishment-at-sea system is anticipated to operate. To make this criteria comparable to other systems which may operate at a different tension the tension difference referred to above should be divided by the desired value of tension to be maintained constant. Thus, on this basis, the performance criteria would be $\frac{\Delta T}{T}$ where ΔT is the difference between the extreme tension qualities encountered in the most severe environment under which the replenishment-at-sea system is intended to operate. Justification of this type of performance criteria is that a certain minimum tension must be maintained to adequately suspend the cargo load while a maximum tension must not be exceeded to prohibit the wire rope from parting. If the difference tetween these two extreme tension qualities can be minimized, the maximum tension encountered can be reduced. Thus larger cargo loads could be carried by replenishment-at-sea rigs utilizing smaller wire rope with smaller associated fittings as compared with equal capacity systems with a larger difference between these two extreme tension values. Relative to performance evaluation, power consumption appears secondary to the capability of maintaining constant tension. However a limitation of the peak power requirement should be considered in the comparison of different systems.

VII. CLOSURE

Subsequent to the preparation of the main body of this report the following has become apparent.

Improvement could be realized if the tension quality controlled was the highest tension value of the two wire ropes used instead of controlling the sum of the tension qualities ($T_a + T_b$) as evolved earlier in this report. Considering cargo trolley transit from the delivering ship to the receiving ship the inboard whip tension (T_a) would be maintained constant while the trolley travels from the delivering ship to mid-span where the tension qualities become equal ($T_b = T_a$). As the trolley passes mid-span enroute to the receiving ship the transfer whip tension becomes greater than the inboard whip tension ($T_b > T_a$), thus the tension quality then controlled is that of the transfer whip (T_b). Using the same comparison as was previously used to establish the superiority of controlling the tension sum constant the cargo load capability (W) becomes 264 lbs. vice 245 lbs. The inboard whip tension difference ($\triangle T_a$) during transit increases from 37.3 lbs. to 55.1 lbs. while the same quality in the transfer whip increases from 37.3 lbs. to 60.6 lbs. The increase in cargo load carrying capability makes controlling the higher tension quality more attractive than controlling the sum

of the tension qualities in spite of increases in the tension differences during cargo transit in both whips. Table 1 offers comparison with the tabulation presented in figure 2 for the control of other tension qualities:

Table I

Region		Controlled Constants T_a or T_b = 330 lbs.
"1"	T_a	330 lbs
	T_b	269.4
	θ_a	26°-08'
"2"	$T_a = T_b$	330
	$\theta_a = \theta_b$	11°-32'
"3"	T_a	274.9
	T_b	330
	θ_b	23°-35'
	ΔT_a	55.1 lbs.
	ΔT_b	60.6
	W	264

The control mechanism shown in figure 1 would require modification so that only the greater of the two wire rope tension values would displace the servo valve spool. The spring rate as evolved in Appendix E would be

reduced from 478 lbs./in. to 257 lbs./in. Figure 5 illustrates a possible differential actuators to servo valve spool arrangement for utilizing the greater wire rope tension value only.

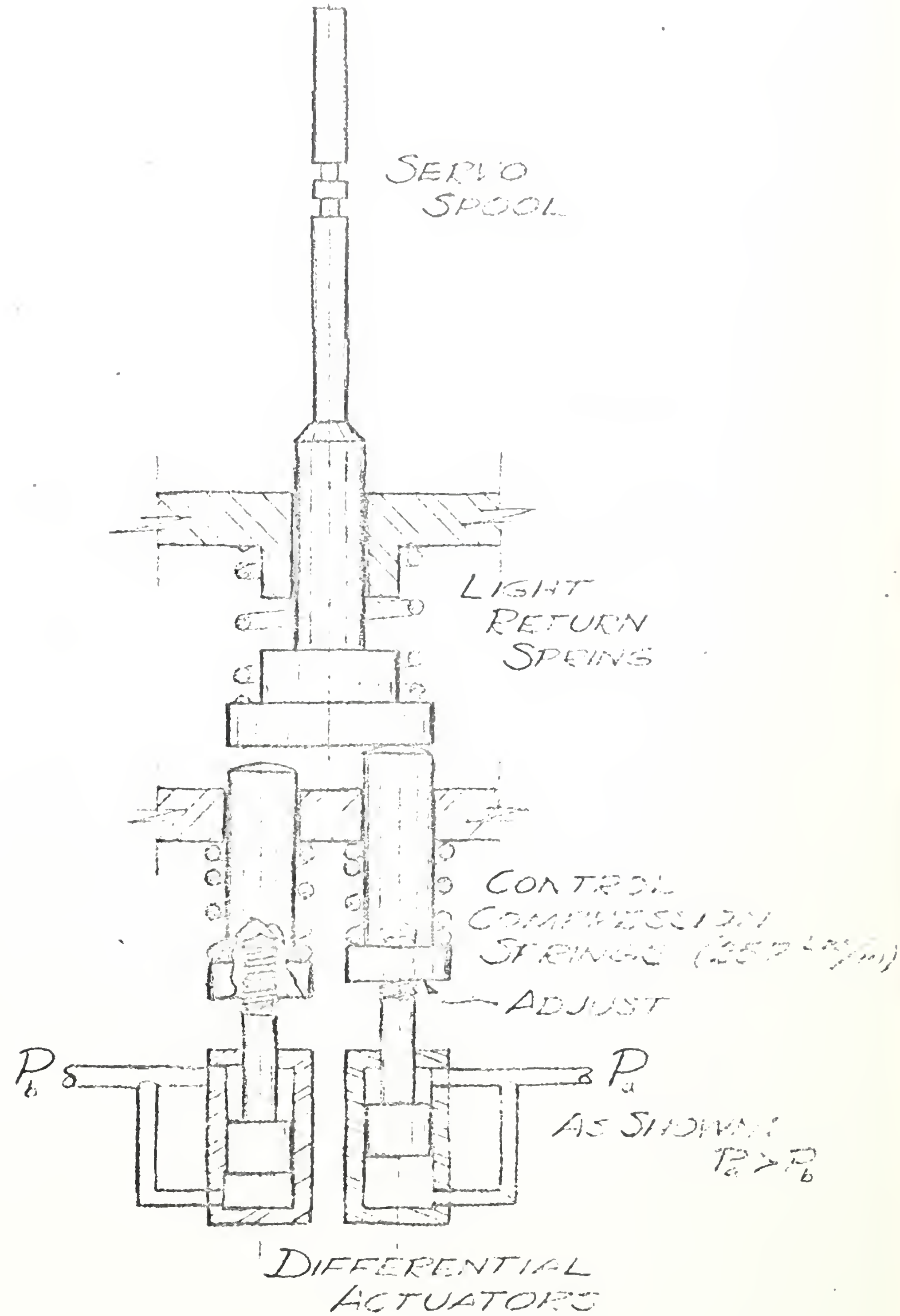


FIGURE 5
CONTROL SYSTEM MODIFICATION

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IX. APPENDICES

APPENDIX A

FORMULATION OF PROBLEM

FORMULATION OF PROBLEM

Figure 1(a) shows a quantity of wire rope; length L_o with a force, tension T_o, applied axially along this wire rope. One end of the wire rope is attached at a fixed point A while the other end, B, is located by the force, Tension T_o.

If the Tension T_o is increased from its initial value T_o to T_1 the wire rope is strained such that the end of the wire rope at point B_o moves to point B_1 and the length is elongated from L_o to L_1, figure 1(b).

If the Tension T_o is decreased from its initial value T_o to T_2 the strain is reduced such that the end of the wire rope has moved to point B_2 and the length is reduced from L_o to L_2, figure 1(c).

Note that the length of the wire, L, varies directly as the tension T applied to the wire,

$$L = KT \tag{1}$$

Compliance with Hooke's Law is assumed: the strain is proportional to the stress.

Suppose that a fixed segment of wire is attached between two points A and B; however the separation between these points is not constant.

For purposes of this illustration point A will be fixed while point B moves. See figure 2.

Let M = separation.

As point B moves to the right the separation M increases from M_o to M_1; the wire rope is elongated an equal amount:

$\Delta L_{o1} = \Delta M_{o1}$ (disregarding catenary effects and the weight of the wire rope) thus L_o increases to L_1.

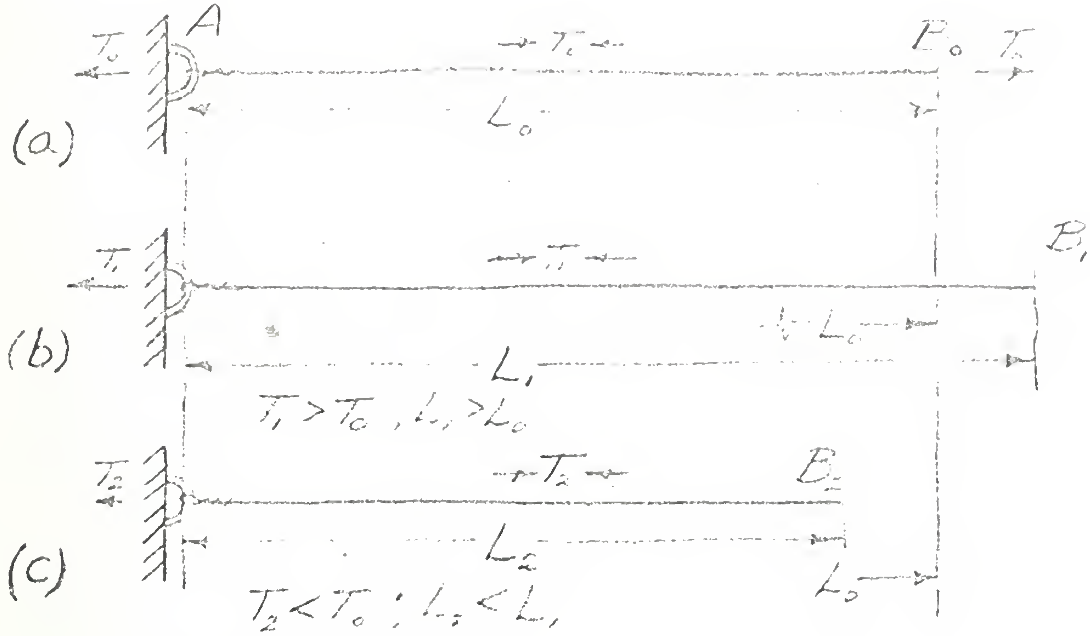


FIGURE 1

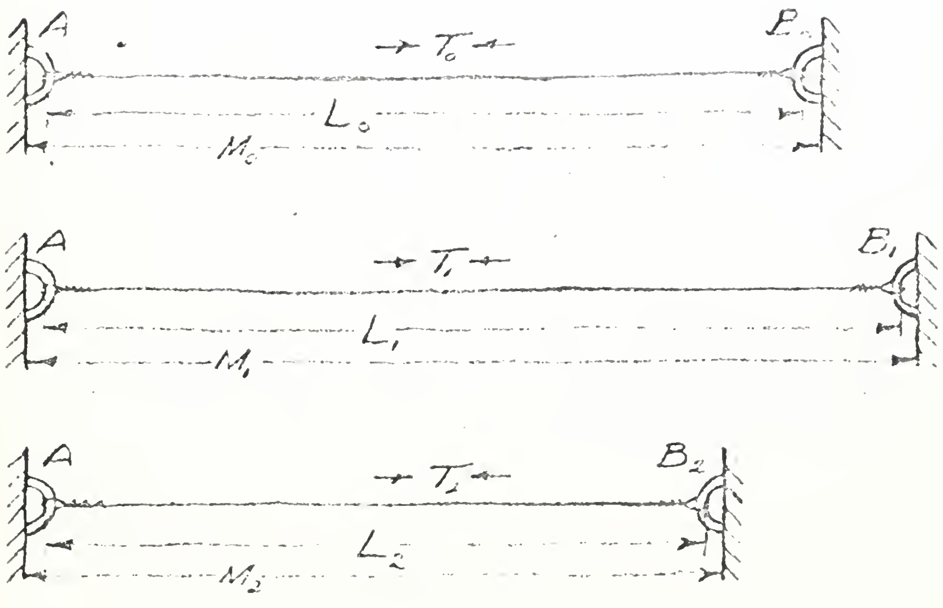


FIGURE 2

To elongate the wire rope an additional tensile load is applied to the wire rope as point B moves to the right. Thus the tension in the wire rope increases from T_0 to T_1.

Similarly if point B moves to the left the separation is decreased from M_0 to M_2, the length of the wire rope is decreased from L_0 to L_2 and the tension in the wire rope decreases from T_0 to T_2.

Now suppose that it is desired to maintain the tension in the wire rope at an optimum value while the wire rope is suspended in a taut manner between two points, the separation of these two points being variable.

To accomplish this the quantity of wire rope suspended between the two terminal points will be varied in response to the variation of separation between these two points.

One end of the wire is attached to a power operated reel, winch drum, in the vicinity of point A. The other end of the wire, the free bitter end, is attached at point B.

For illustrative purposes, consider point A to be at a fixed point while point B moves. (This is only a manner of viewing this situation, the important matter is not which point moves, but that there is relative motion between these points and that the distance between these two points varies.) See figure 3.

In figure 3(a), as an initial condition, assume that the wire rope segment suspended between the winch drum at A, and the point B has a length L_0 and that the tension corresponding to the length L_0 of the quantity of wire rope suspended between points A and B is T_0.

In figure 3(b), point B has moved from B_0 to B_1 and the length of the wire rope suspended between points A and B has been elongated from L_0 to L_1 an increment represented by ΔL. The tension in the wire has increased from T_0 to T_1.

In figure 3(c), point B remains at B_1 and the length of the wire rope segment remains L_1 as in figure 3(b). However an additional quantity of wire rope, corresponding to the increment $\triangle L$ has been added by rotation of the winch drum in a clockwise direction. The tension T_1 of figure 3(b) is decreased to the desired value T_o.

In figure 3(d), point B has moved from B_o to B_2 (referring back to figure 3(a)) and the length of the wire rope suspended between points A and B has been shortened from L_o to L_2, an increment represented by $\triangle L$. The tension in the wire has decreased from T_o to T_2.

In figure 3(e), point B remains at B_2 and the length of the wire rope segment suspended between points A and B remains L_2 as in figure 3(d). However, an increment of wire rope, corresponding to $\triangle L$ has been taken from the quantity of wire rope suspended between the points A and B by rotation of the winch drum in a counter clockwise direction. The tension T_2 of figure 3(d) is increased to the desired value T_o.

Examination of figure 4 shows three items which influence tension T in the wire rope segment between points A and B.

These three items are:
(a) Movement of point A, x
(b) Movement of point B, y
(c) Rotation of the winch drum, z

For convenience, consider these items to be positive if they tend to increase the tension load T of the wire. Thus:

x is positive if A moves to the left.
y is positive if B moves to the right.
z is positive if the winch drum rotates counter clockwise. Consider z to be the movement of the wire caused by the rotation of the winch drum.

If the wire rope is initially placed between points A and B in a manner as illustrated in figure 4 with a resultant tension T_o. Then points A and B move and the winch drum rotates the resultant change of tension T may be expressed:

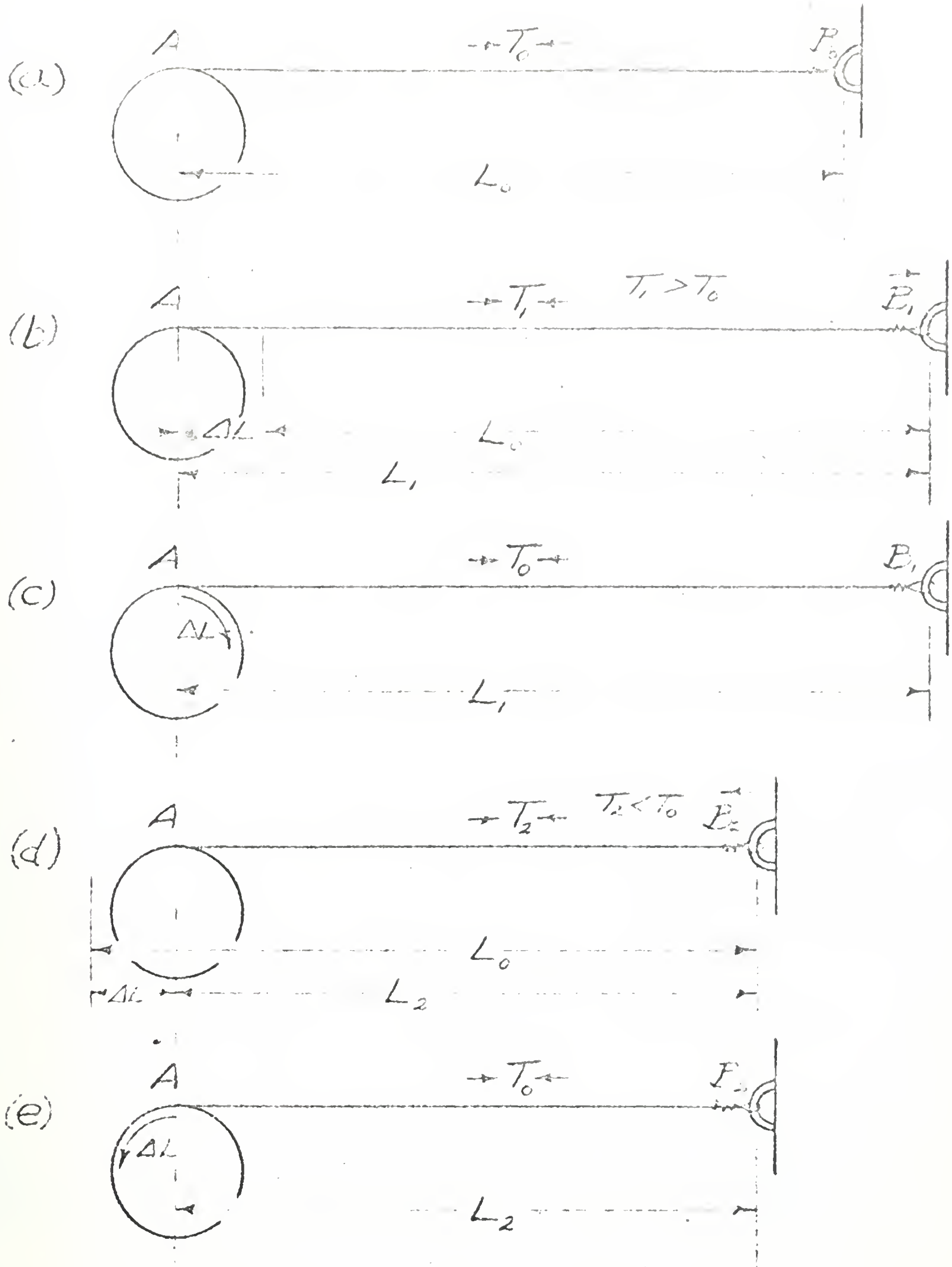


FIGURE 3

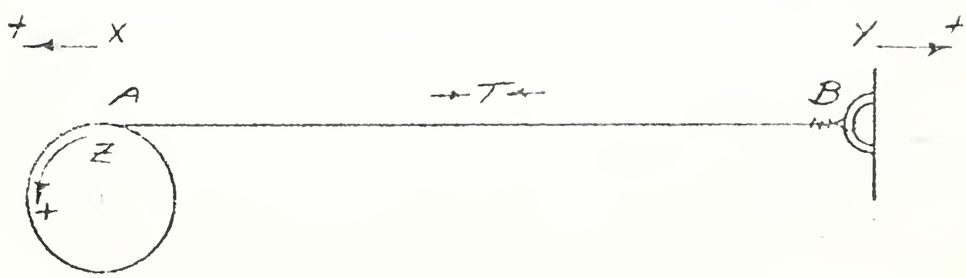


FIGURE 4

$$\Delta T \sim \Delta x + \Delta y + \Delta z \tag{2}$$

or

$$K \Delta T = \Delta x + \Delta y + \Delta z \tag{3}$$

Considering time:

$$K \dot{T} = \dot{x} + \dot{y} + \dot{z} \tag{4}$$

Ideally, if T is to be held constant:

$$K \dot{T} = 0 = \dot{x} + \dot{y} + \dot{z} \tag{5}$$

The quantity $\dot{x} + \dot{y} + \dot{z}$ may be split into two sub-quantities readily identifiable with replenish-at-sea.

Sub-quantity $\dot{x} + \dot{y}$ is the separation variation between the terminal or suspension points of the replenish-at-sea rig. This sub-quantity itself is the sum of the movements (velocities) of the rig suspension point at each replenishing ship. $\dot{x}$ is predominantly due to the roll of the delivering ship while $\dot{y}$ is predominantly due to the roll of the receiving ship as illustrated in figure 5.

Sub-quantity $\dot{z}$ is the rotation (peripheral velocity) of the winch drum.

To maintain constant tension, equation (5) should be continuously satisfied during the replenishment-at-sea operation. Thus sub-quantity $\dot{x} + \dot{y}$ is the disturbance while sub-quantity $\dot{z}$ is the correctional response intended to maintain the tension constant.

Because $\dot{z}$ follows $\dot{x} + \dot{y}$ it is not possible to maintain the value of tension T ideally constant; it should be possible to maintain the tension within close tolerances however.

The arrangement shown in figure 5 serves to illustrate the manner in which tension is to be controlled. However, the method of transferring cargo from ship to ship while simultaneously controlling tension has not been illustrated.

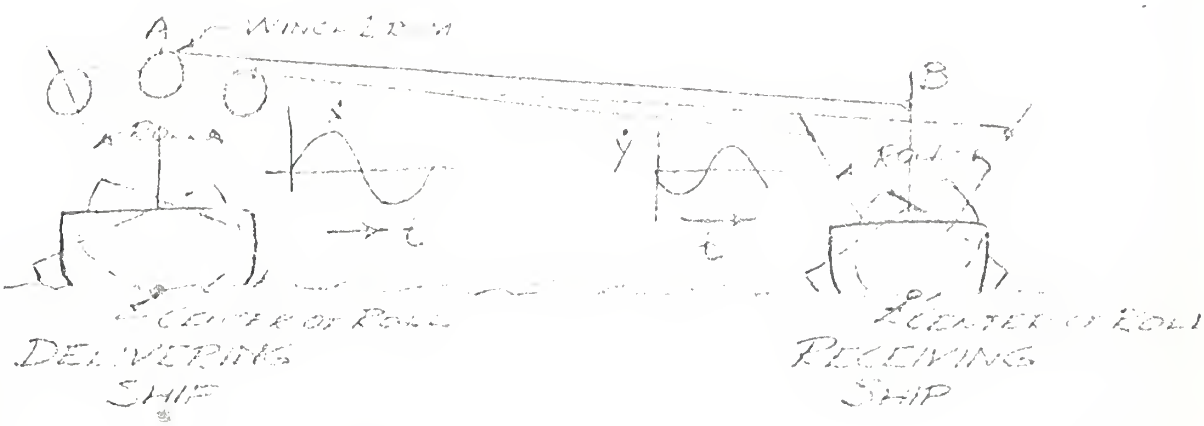


FIGURE 5

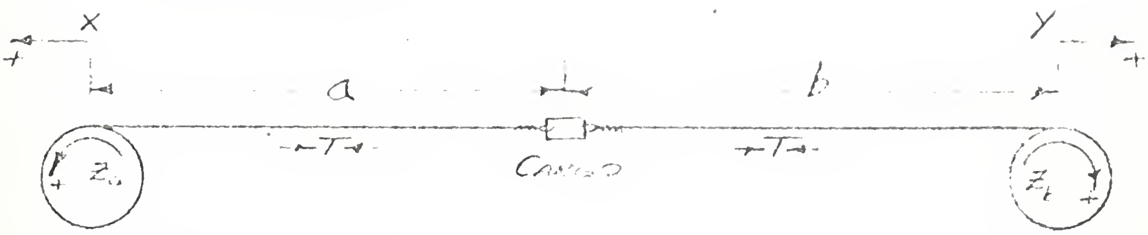
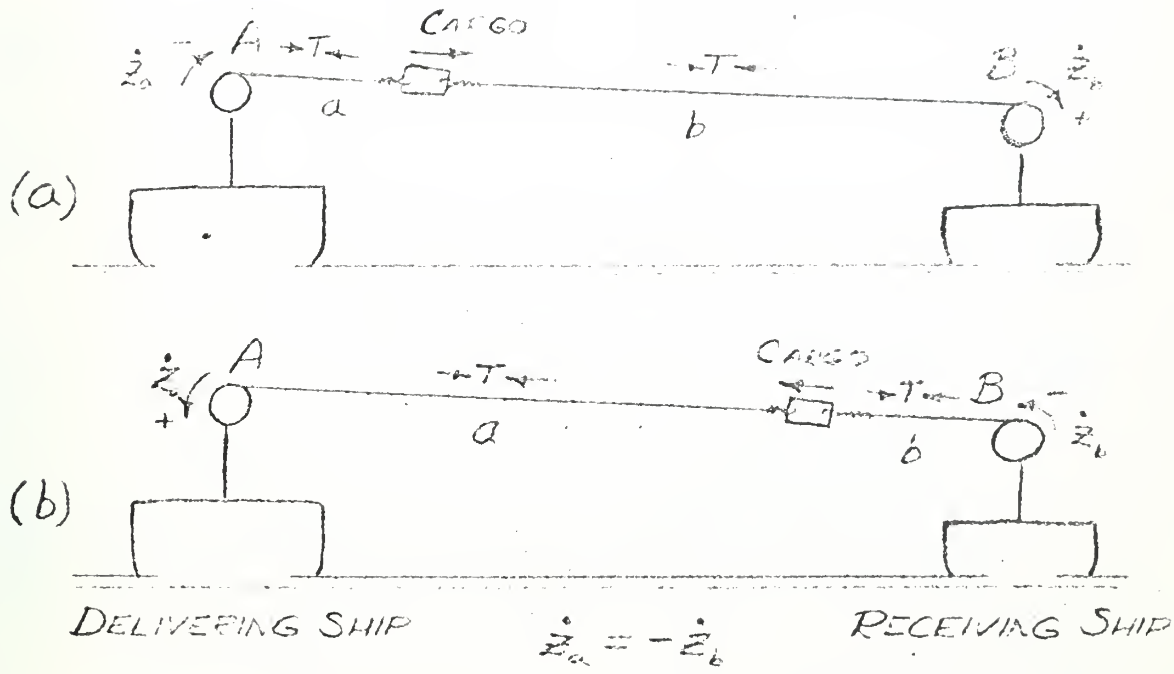


FIGURE 6



(a) ROTATION OF BOTH WINCH DRUMS CLOCKWISE, CARGO MOVES FROM A TO B.

(b) ROTATION OF BOTH WINCH DRUMS COUNTER-CLOCKWISE, CARGO MOVES FROM B TO A.

FIGURE 7

For initial illustrative purposes, referring back to figure 5; suppose that the bitter end of the wire rope shown in figure 5 is detached from point B and attached to a cargo load. In addition a winch drum is placed aboard the receiving ship in the vicinity of point B. An additional quantity of wire rope is attached to the winch drum at point B; the bitter end of this added quantity of wire rope is attached to the cargo load. The resulting rig configuration is shown in figure 6.

Consider the cargo to be weightless for initial explanatory purposes.

Referring back to equations (4) and (5), note that the following substitution can be made and that these basic relationships remain applicable:

$$z = z_a + z_b \tag{6}$$

where:

z_a is positive if the winch drum at A rotates counterclockwise.

z_b is positive if the winch drum at B rotates clockwise.

Equations (4) and (5), rewritten are:

$$K\dot{T} = \dot{x} + \dot{y} + \dot{z}_a + \dot{z}_b \tag{7}$$

$$K\dot{T} = 0 = \dot{x} + \dot{y} + \dot{z}_a + \dot{z}_b \;;\; (\text{for } \Delta T = 0) \tag{8}$$

Note also that the position of the cargo is related to z_a and z_b. With initial conditions that $z_a = 0$ when the cargo is at point A and $z_b = 0$ when the cargo is at point B the following relationships apply:

$$a = -z_a \tag{9}$$

$$b = -z_b \tag{10}$$

with

a = distance from point A to the cargo.

b = distance from point B to the cargo.

The reason for the minus signs is due to the convention established through which z_a and z_b are positive if they result from rotation of the winch drums tending to increase the tension in the wire rope quantities.

Thus:

$\dot{z}_a = -\dot{a}$ represents the velocity of the cargo toward point A, the delivering ship.

$\dot{z}_b = -\dot{b}$ represents the velocity of the cargo toward point B, the receiving ship.

If $\dot{x} = \dot{y} = 0$ or if $\dot{x} + \dot{y} = 0$ and tension T is maintained constant, equation (8) can be reduced to:

$$\dot{z}_a + \dot{z}_b = 0 \qquad (11)$$

The conditions established by equation (11) represent pure transfer of a weightless load from A to B or B to A without tension change.

Thus to transfer the cargo load the winch drums are rotated in the same directions, both clockwise or both counter clockwise, as depicted in figure 7.

Up to this point the cargo has been ideally considered weightless. Now consider that the cargo realistically has weight. It is apparent that tension T now serves the useful purpose of suspending the cargo in space. With this the practical limitations of tension T evolve, thus:

Tension T must be of a sufficient quantity to safely suspend the cargo load clear of the ship structure and clear of the sea during the transfer of a cargo load from ship to ship <u>but</u> the tension T must also be kept sufficiently low so that no hazard of parting (tensile failure) of the wire rope exists.

Figure 8 is a free body diagram illustrating how the cargo load is suspended by tension.

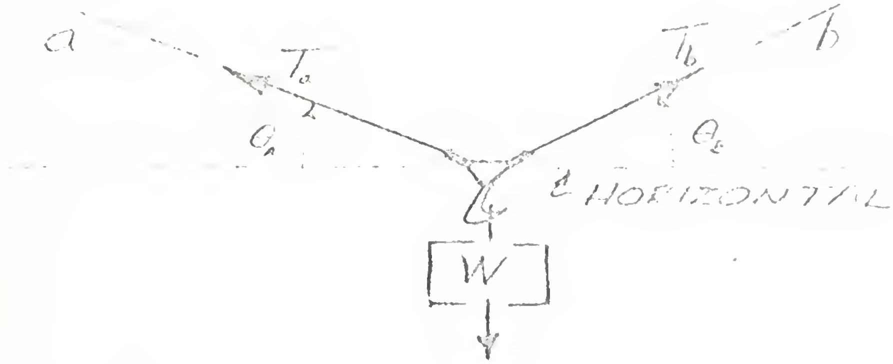


FIGURE 8

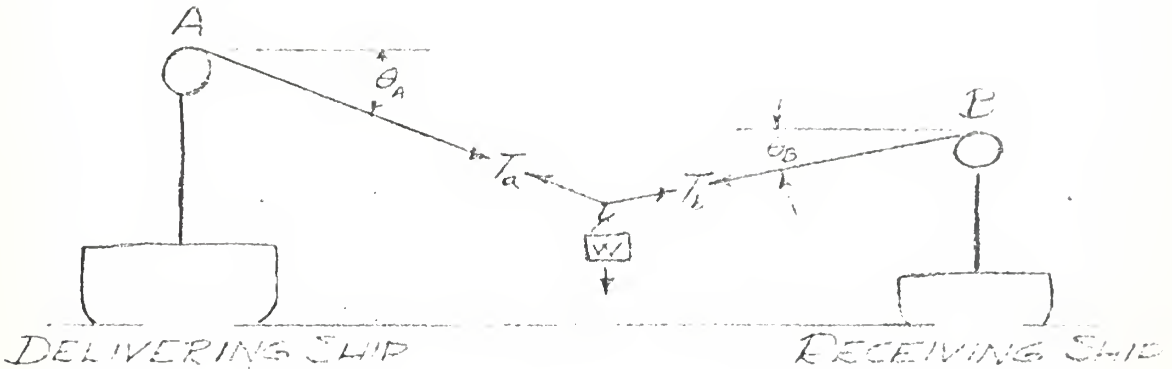


FIGURE 9

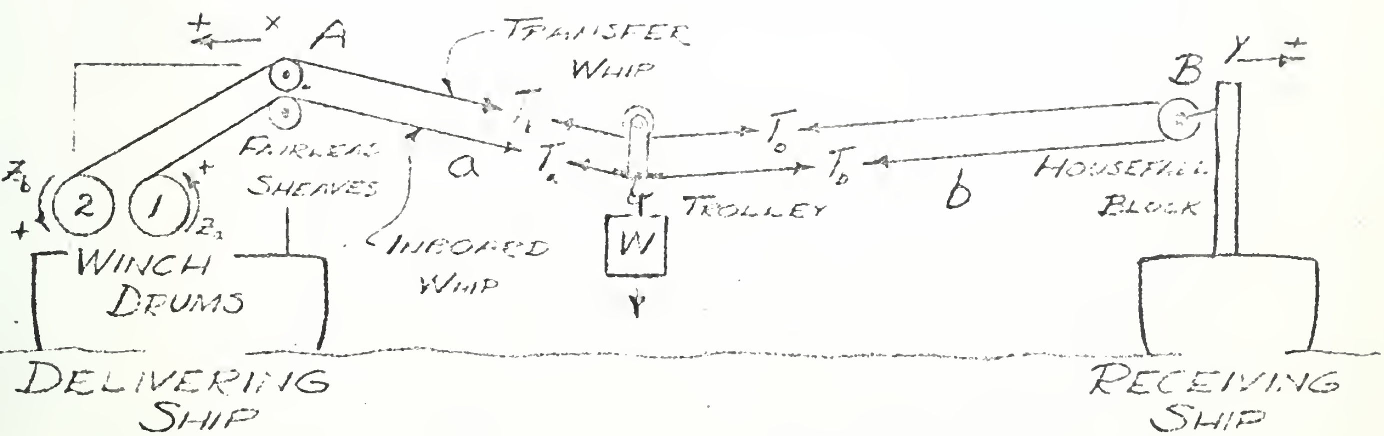


FIGURE 10

From figure 8:

$$W = T_a \sin\theta_A + T_b \sin\theta_B \tag{12}$$

$$T_a \cos\theta_A = T_b \cos\theta_B \tag{13}$$

where:

W = Load weight, lbs.

T_a = Tension, wire rope segment a.

T_b = Tension, wire rope segment b.

θ_A = Angle of wire rope segment a from horizontal.

θ_B = Angle of wire rope segment b from horizontal.

Note that the tension T_a does not normally equal the tension T_b. By limiting the weight carried, the required values of T_a and T_b are nearly equal. Expressed mathematically:

$$T_a \rightarrow T_b \rightarrow \bar{T} \text{ as } \frac{W}{\bar{T}} \rightarrow 0 \tag{14}$$

Where $\bar{T}$ = the desired mean value of tension.

Considering cargo weight, figure 7 is redrawn; see figure 9.

The most desirable manner of controlling the movement of the cargo load is that this movement be controlled with reference to the delivering ship, point A, while the cargo load is in the vicinity of the delivering ship and with reference to the receiving ship, point B, while the cargo load is in the vicinity of the receiving ship. During the portion of the cargo transit, in which the cargo load is sufficiently clear of both ships it is immaterial whether the cargo movement is controlled relative to the delivering ship or relative to the receiving ship.

Control of cargo movement relative to the delivering ship (point A): The winch drum aboard the delivering ship, at point A, is rotated corresponding to the speed at which it is desired to move the cargo relative to A; the winch drum aboard the receiving ship, at point B, is rotated as necessary to maintain constant tension, T_b = constant.

Control of cargo movement relative to the receiving ship (point B): The winch drum aboard the receiving ship, at point B, is rotated corresponding to the speed at which it is desired to move the cargo relative to B; the winch drum aboard the delivering ship, at point A, is rotated as necessary to maintain constant tension, T_a = constant.

In figure 10 the arrangement shown in figure 9 has been modified principally by moving the winch drum which was shown aboard the receiving ship, figure 9, to the delivering ship as shown. At B is placed a sheave (roller or pulley). The wire rope segment extending from the cargo toward the receiving ship is led through a 180° bend about the sheave at the receiving ship and returns to the cargo where it passes under a sheave located above the cargo load and the attachment points of the wire rope segments to the load supporting trolley. From the trolley this wire rope segment continues over a fairlead sheave to the second winch drum now located aboard the delivering ship. An additional fairlead sheave has also been placed aboard the delivering ship to support the wire rope segmemt from the winch drum which was initially aboard the delivering ship to the cargo trolley. Point A is now defined as the location of the fairlead sheaves aboard the delivering ship and essentially is the point aboard the delivering ship from which the cargo transfer rig, extending toward the receiving ship, is suspended.

For convention, the wire rope quantity wound about winch drum (1), led over the lower fairlead sheave at point A and attached to the trolley on the left side as depicted in figure 10 is called the INBOARD WHIP, (the inboard whip is also indicated by tension designation T_a in figure 10). The wire rope quantity wound about winch drum (2) led over the upper fairlead sheave, under the trolley sheave, about the housefall sheave at point B and back to the trolley where it is attached to the right side of the trolley as depicted in figure 10 is called the TRANSFER WHIP, (the transfer whip is indicated by tension designation T_b in figure 10).

The replenishment-at-sea rig shown in figure 10 with the use of tensioning control is called the TENSIONED MODIFIED HOUSEFALL rig.

From a control and operational viewpoint the Tensioned Modified Housefall rig has two major advantages has compared to the replenishment-at-sea rig shown in figure 9 (Tensioned Burtoning).

1. With both winch drums aboard the delivering ship, simultaneous control and drive of these two winch drums are greatly facilitated as compared to the arrangement depicted in figure 9 where one winch drum is located on each ship.

2. With both winch drums aboard the delivering ship the requirement to place a winch drum aboard the receiving ship is eliminated. Therefore a great economy results relative to the incorporation of the Tensioned Modified Housefall replenishment-at-sea method. Only the delivering ships, with a primary mission of replenishment-at-sea, require extensive installations. Thus any ship is a potential "receiving ship." (The designation "receiving ship" specifies the normal role of the ship not configured to be the "delivering ship", cargo flow can be in either direction.)

The matter of placing the cargo upon the trolley at the delivering ship and removing the cargo from the trolley at the receiving ship is dependent upon other machinery. At this point, accept the fact that it is possible to load or unload cargo to or from the trolley at either ship.

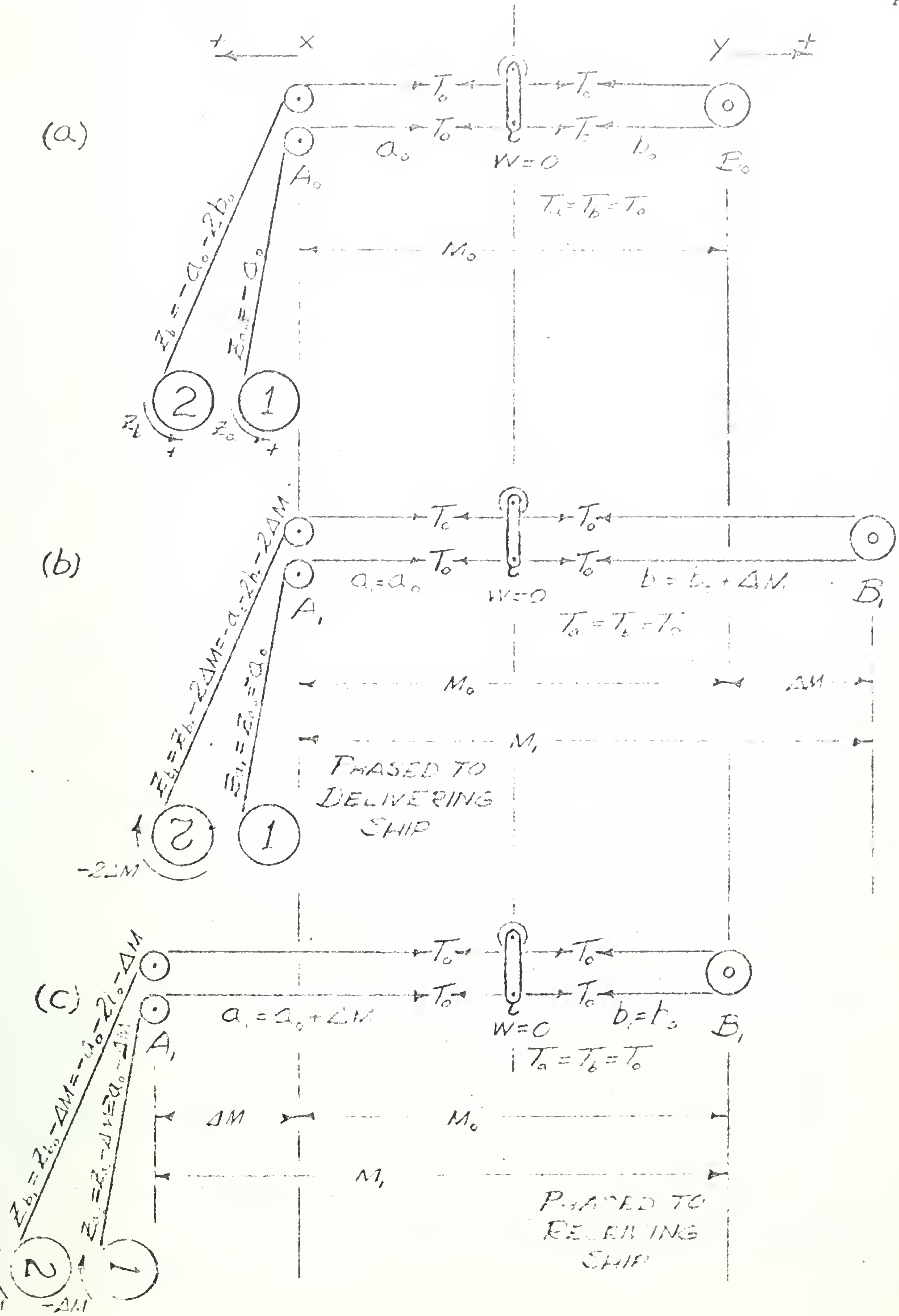
Equations previously derived are modified as necessary and rewritten below. As before, it is convenient to start with a weightless cargo load.

For $W = 0$, $T_o = T_a = T_b$ = constant. The value of tension T_o described here and depicted in figure 11 corresponds to half the previous value represented in figures 8 and 9 as twice as many parts of wire rope now extend between the two ships and suspend the cargo load.

With $T_o = T_a = T_b$ = Constant, with $W = 0$

$$K\dot{T} = \dot{x} + \dot{y} + \dot{z}_a + \dot{z}_b = (0 \text{ for } \Delta T = 0) \qquad (7) \ \& \ (8)$$

z_a is positive if winch drum (1) rotates counterclockwise as depicted in figure 11.



(a)
X
Y
T_o
a_o
W=0
b_o
A_o
B_o
$T_a = T_b = T_o$
M_o
$Z_b = -a_o - 2b_o$
$Z_a = -a_o$
2
1
(b)
$a_1 = a_o$
W=0
$b = b_o + \Delta M$
A_1
B_1
$T_a = T_b = T_o$
M_o
ΔM
M_1
PHASED TO
DELIVERING
SHIP
$-2\Delta M$
(c)
$a_1 = a_o + \Delta M$
W=0
$b_1 = b_o$
A_1
B_1
$T_a = T_b = T_o$
ΔM
M_o
M_1
$-\Delta M$

FIGURE 11

z_b is positive if winch drum (2) rotates counterclockwise as depicted in figure 11.

With point A as the reference point relative to the quantities of wire rope used in the rig:

$$a = -z_a \qquad (9)$$

$$a + 2b = -z_b \qquad (15)$$

Hence:

$$b = \frac{z_a - z_b}{2} \qquad (16)$$

Differentiating with respect to time:

$$\dot{a} = -\dot{z}_a \qquad (17)$$

$$\dot{b} = \frac{\dot{z}_a - \dot{z}_b}{2} \qquad (18)$$

Where:

$\dot{a}$ = the velocity of the trolley <u>away</u> from point A, the delivering ship.

$\dot{b}$ = the velocity of the trolley <u>away</u> from point B, the receiving ship.

As before, if $\dot{x} = \dot{y} = 0$ or if $\dot{x} + \dot{y} = 0$, $W = 0$ and T_o is held constant:

$$\dot{z}_a + \dot{z}_b = 0 \qquad (11)$$

Again: The conditions established by equation (11) represent pure transfer of a weightless load from A to B or B to A without tension change.

To transfer the cargo load the winch drums are rotated in opposite directions at the same speed with reference to figure 11.

<u>PHASED TO:</u>

For convenience the term "phased to" is introduced as related to the Tensioned Modified Housefall method of replenishment-at-sea.

<u>Phased to the Delivering Ship</u>: Trolley transfer is controlled relative to the delivering ship. The velocity $\dot{a}$ is controlled. Tension corrections are made in such a manner so as not to affect $\dot{a}$.

<u>Phased to the Receiving Ship</u>: Trolley transfer is controlled relative to the receiving ship. The velocity $\dot{b}$ is controlled. Tension corrections are made in such a manner so as not to affect $\dot{b}$.

Figure 11 illustrates tensioning control phased to the delivering ship with zero trolley transfer speed, $\dot{a} = 0$, is figure 11(a) the arrangement is illustrated with a separation $= M_0$. In figure 11(b) the separation has increased by ΔM to M_1.

Since the separation change ΔM affects two parts of wire rope the correction required is $2\Delta M$. In figure 11b note that a correction has been applied with the transfer whip, winch drum (2). Mathematically the separation, variation and tension correction could be represented:

$$KT_0 = \underbrace{x_0 + y_0}_{\pm 2\Delta M} + z_{a_0} + \underbrace{z_{b_0}}_{\mp 2\Delta M} \qquad (19a)$$

Separation variation $\rightarrow \pm 2\Delta M$ $\qquad$ $\mp 2\Delta M \leftarrow$ Tension Correction

$$KT_0 = \overbrace{x_1 + y_1}^{\parallel} + z_{a_0} + \overbrace{z_{b_1}}^{\parallel} \qquad (19b)$$

Note that the quantity $z_a \; (= -a)$ is not altered from equation (19a) to (19b) which indicates no change in the position of the trolley as related to point A, the delivering ship.

Figure 11 also illustrates tensioning control phased to the receiving ship with zero trolley transfer speed, $\dot{b} = d$. In figure 11(a) the arrangement is illustrated with a separation $= M_0$. In figure 11(c) the separation has increased by ΔM to M_1.

In figure 11(c) note that the correction $2\Delta M$ is applied by both the Inboard Whip and the Transfer Whip, winch drums (1) and (2), an increment ΔM each. Mathematically the separation variation and tension correction could be represented:

$$KT_o = x_o + y_o + z_{a_o} + z_{b_o} \tag{19a}$$

separation variation ... $\pm 2\Delta M \quad (\mp \Delta M \quad \mp \Delta M)$ — tension correction

$$KT_o = x_1 + y_1 + z_{a_1} + z_{b_1} \tag{19c}$$

In which:

$$z_{a_1} = z_{a_o} \mp \Delta M$$

$$z_{b_1} = z_{b_o} \mp \Delta M$$

Substituting in equation (16):

$$b_o = \frac{z_{a_o} - z_{b_o}}{2}$$

$$b_1 = \frac{z_{a_1} - z_{b_1}}{2} = \frac{(z_{a_o} \mp \Delta M) - (z_{b_o} \mp \Delta M)}{2}$$

$$\therefore\ b_1 = \frac{z_{a_o} - z_{b_o}}{2} = b_o$$

thus the relationship $\dfrac{z_{a_o} - z_{b_o}}{2}$ (= b) is preserved from equation (19a) to (19c) which indicates no change in the position of the trolley as related to point B, the receiving ship.

The examples of tensioning control with phasing cited above concern ideal conditions where the trolley speed ($\dot{a}$ or $\dot{b} = 0$) is zero and the cargo weight ($W = 0$) is also zero. These conditions are only for illustrative convenience. The trolley speed ($\dot{a}$ or $\dot{b}$) need not be zero. Trolley movement may be zero speed (stopped), a constant speed, a constant acceleration or any convenient function of time. The cargo may have weight. It is still possible to control the trolley movement related to A or B, (i.e.: phased to the delivering ship or phased to the receiving ship) without tensioning variation or correction affecting this relationship.

APPENDIX B

TENSION VARIATION AND CORRECTION WITH PHASING: TENSIONED MODIFIED HOUSEFALL

(Appendix A of SIXTH REPORT, ULTIMATE TRANSFER RIG . . . March 1963 by author)

TENSION VARIATION AND CORRECTION WITH PHASING; TENSIONED MODIFIED HOUSEFALL

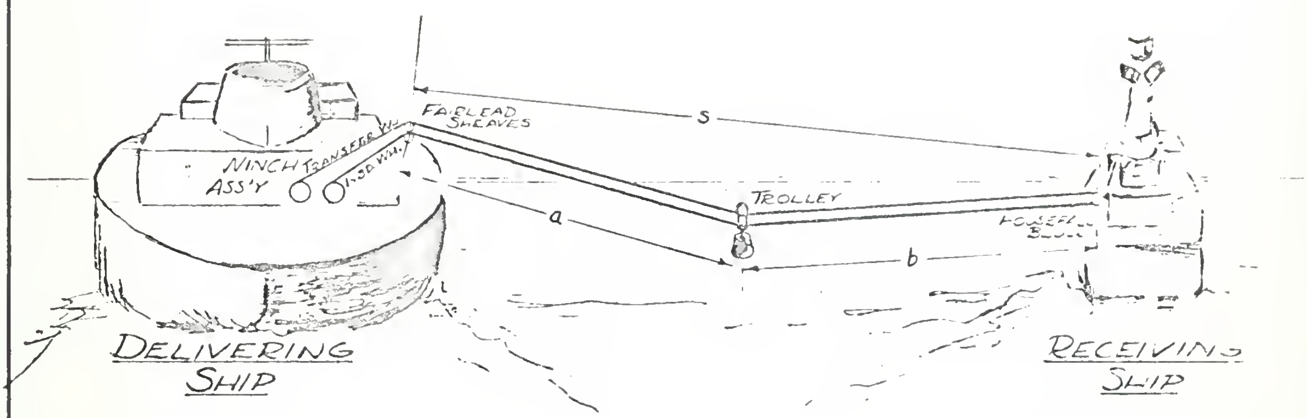
For illustrative purposes, the trolley is at a fixed location as related to the transfer operation, the trolley is located at some point along its movement between the replenishing ships and no control evolution is taking place to move the trolley from one ship to the other. This discussion concerns tension changes away from the desired value of tension, the subsequent action to regain the desired value of tension, and the effects of tension discrepancy and correction to the distance of the trolley along the wire to either of the replenishing ships.

The first portion of this discussion concerns the effect of tension discrepancy upon the distance of the trolley along the wire to either of the replenishing ships. The second and third portions concern the effect of tension correction upon the distance of the trolley to either of the replenishing ships dependent upon phasing. The second portion concerns tension correction phased to the Delivering Ship. The third portion concerns tension correction phased to the Receiving Ship. The basic rig with terminology for this discussion is illustrated in figure A-1.

A-1 Tension Discrepancy.

As transfer control operation is not included in this discussion, the only item affecting rig tension is variation of the distance between the suspension points of the rig. If "S" varies and no wire is added to or taken from the rig the trolley does not move along the wire toward either replenishing ship. The trolley is fixed at distance "a" from the upper tangent of the Inboard Whip fairlead sheave as the length of the Inboard Whip is maintained constant. The Transfer Whip within the rig is equal to a distance "a + 2b" The Transfer Whip is held taut by tension and confined in its length from the Receiving Ship to the trolley, by the structure of the trolley, parallel to the Inboard Whip. As the trolley confines the upper part of the Transfer Whip to the Receiving Ship and the bitter end of the Transfer Whip is attached to the trolley, tension divides this bight of the Transfer Whip to two equivalent parts each equal to "b" . The distance from the trolley to the replenishing ships measured along the wire "a" and "b" , does not alter as tension varies.

With a change in tension the trolley moves in an arc of radius "a" centered at the upper tangent of the Inboard Whip fairlead sheave aboard the Delivering Ship and the trolley also moves in an arc of radius "b"



a = DISTANCE; DELIVERING SHIP TO TROLLEY ALONG WIRE

b = DISTANCE; RECEIVING SHIP TO TROLLEY ALONG WIRE

S = DISTANCE; DELIVERING SHIP TO RECEIVING SHIP MEASURED BETWEEN RIG SUSPENSION POINTS

Δx = INCREMENT OF WIRE PER RIG PART NECESSARY TO CORRECT RIG TENSION TO DESIRED VALUE

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centered at the pin of the housefall block aboard the Receiving Ship. These are relative conditions. To visualize the relative situation it is convenient to confine the trolley to a stationary point thus as tension changes the upper tangent of the fairlead sheave aboard the Delivering Ship moves along an arc of radius "a" centered at the Inboard Whip bitter end attached to the trolley and the pin of the housefall block aboard the Receiving Ship moves along an arc of radius "b" centered at the approximate center point of the trolley structure between the whips. Figure A-2 illustrates the rotation of the rig suspension points about the trolley.

To summarize, the significant item is that tension variations from the desired value of tension do not move the trolley along the wire toward either replenishing ship. So long as sufficient tension is maintained to adequately suspend the cargo load clear of replenishing ship structure and clear of the sea, there is no hazard of collision to the cargo due to separation variation.

A-2 <u>Tension Correction, Phased to Delivering Ship.</u>

Figure A-3 is an exaggerated representation of the effect of tension variations upon trolley location as related to the Delivering Ship. Trolley location "A" is the location of the trolley when tension is at the desired valve. Trolley location "B" is a trolley location with tension increased beyond the desired value and trolley location "C" is a trolley location with tension diminished below the desired value.

To correct to the desired value of tension an increment "Δx" must be added or taken from each part of the rig. With tensioning control phased to the Delivering Ship a length of wire equal to "$2\Delta x$" is added to or taken from the Transfer Whip while no change is made in the Inboard Whip. Thus the distance "a", Delivering Ship to trolley, is not affected by the tension correction as the length of the Inboard Whip is unchanged. The length of the Transfer Whip between the Delivering Ship and the trolley, parallel to the Inboard Whip remains the same. The length of wire added to or taken from the rig affects the portion of the rig between the trolley and the Receiving Ship. This length is halved by the housefall block as the Transfer Whip is in two parts between the trolley and the Receiving Ship.

Location "B" represents the condition where ship separation "S" has increased thereby increasing tension. The trolley moves up along the arc at radius "a" centered at the upper tangent of the Inboard Whip fairlead sheave aboard the Delivering Ship. A length "$2\Delta x$" is added to the Transfer Whip, the desired rig tension is restored and the trolley is returned to location "A" without the distance "a" being altered.

Location "C" represents the condition where ship separation "S" has decreased thereby decreasing tension. The trolley moves down along the

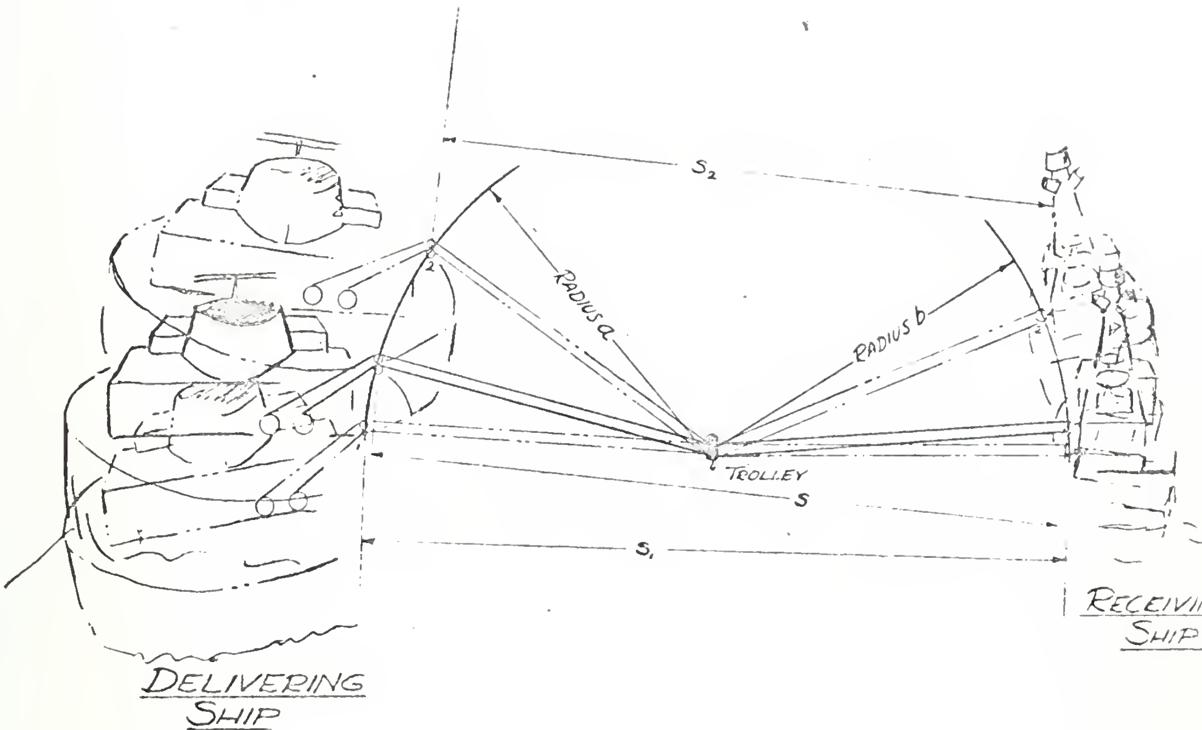

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FIGURE A-2 EFFECTS OF
VARIATION OF
SEPARATION "S"

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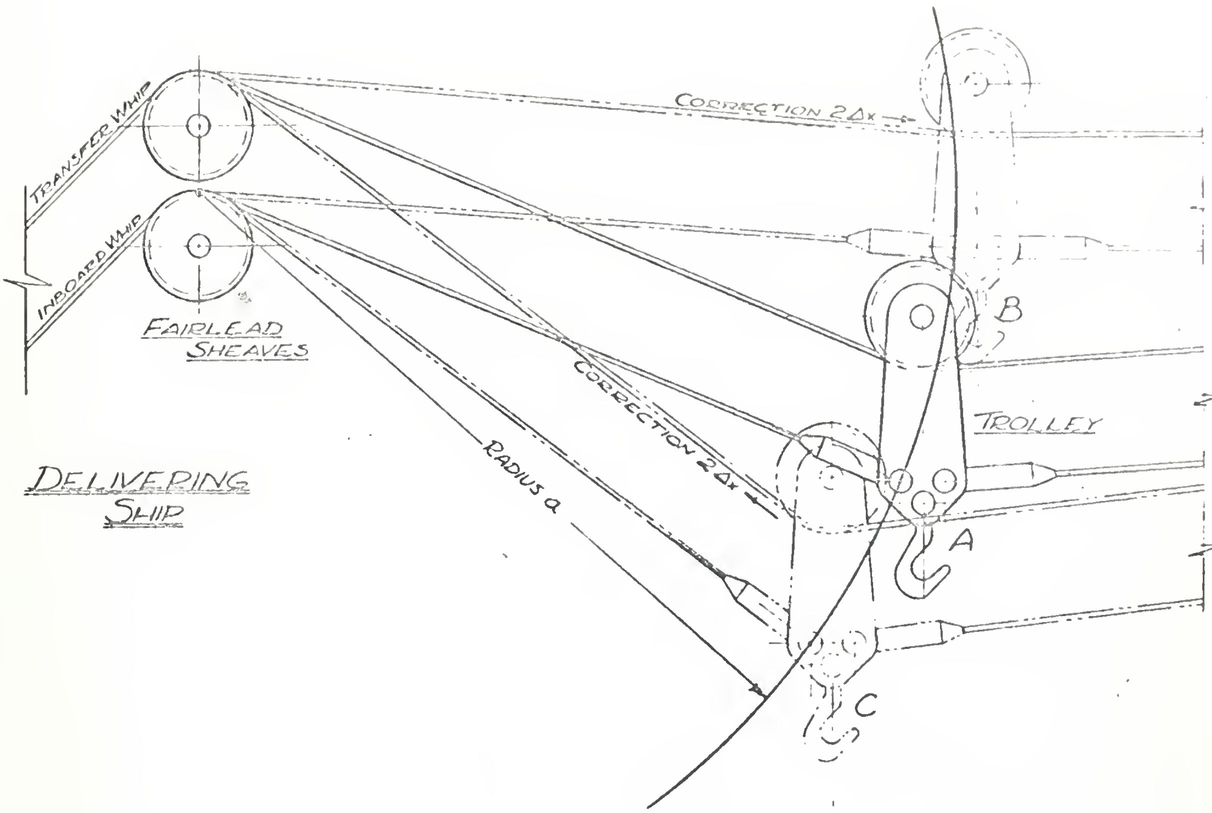

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FIGURE A-3 PHASED TO DELIVERING SHIP, DELIVERING SHIP END

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arc at radius "a" centered at the upper tangent of the Inboard Whip fairlead sheave aboard the Delivering Ship. A length "$2\Delta x$" is taken from the Transfer Whip, the desired rig tension is restored and the trolley is returned to location "A" without the distance "a" being altered.

Figure A-4 illustrates the effects of tension corrections to the lengths of "a" and "b" when phased to the Delivering Ship.

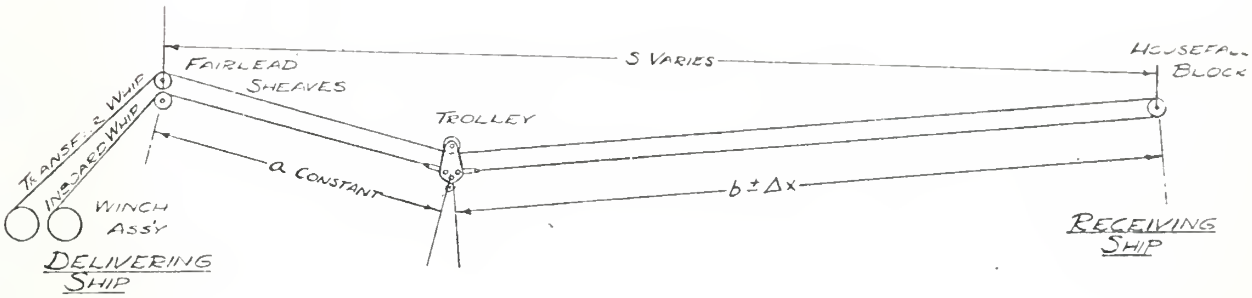
A-3 Tension Correction, Phased to Receiving Ship.

Figure A-5 is an exaggerated representation of the effect of tension variations upon trolley location as related to the Receiving Ship. Trolley location "A" is the location of the trolley when tension is at the desired value. Trolley location "B" is a trolley location with tension increased beyond the desired value, and trolley location "C" is a trolley location with tension diminished below the desired value.

To correct to the desired value of tension an increment "Δx" must be added or taken from each part of the rig. With tensioning control phased to the Receiving Ship a length of wire equal to "Δx" is added to or taken from both the Transfer Whip and the Inboard Whip. A total wire length "$2\Delta x$" is added to or taken from the rig: "Δx" for each whip. The Transfer Whip between the Delivering Ship and the trolley is altered by the increment "Δx" and similarly the Inboard Whip between the Delivering Ship and the trolley is altered by the increment "Δx". Thus the portion of the Transfer Whip extending from the Delivering Ship to the trolley is maintained equal to the Inboard Whip. This portion of the Transfer Whip is maintained parallel to the Inboard Whip by the trolley structure. The trolley sheave roller and the housefall sheave do not rotate about their own axes. The Transfer Whip bight extending from the trolley to the housefall block at the Receiving Ship and back to the trolley, where its bitter end is attached, is not altered. Thus the distance "b", Receiving Ship to trolley, is not affected by the tension correction. The lengths of wire added to or taken from the rig affect the portion of the rig between the trolley and the Delivering Ship.

Location "B" represents the condition where ship separation "S" has increased thereby increasing tension. The trolley moves up along the arc at radius "b" centered at the pin of the housefall block aboard the Receiving Ship. Two lengths "Δx" are added: "Δx" to the Transfer Whip and "Δx" to the Inboard Whip. The desired rig tension is restored and the trolley is returned to location "A" without the distance "b" being altered.

Location "C" represents the condition where ship separation "S" has



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			HD. ENGR.		FIGURE A-4 PHASED TO DELIVERING SHIP, ENTIRE RIG
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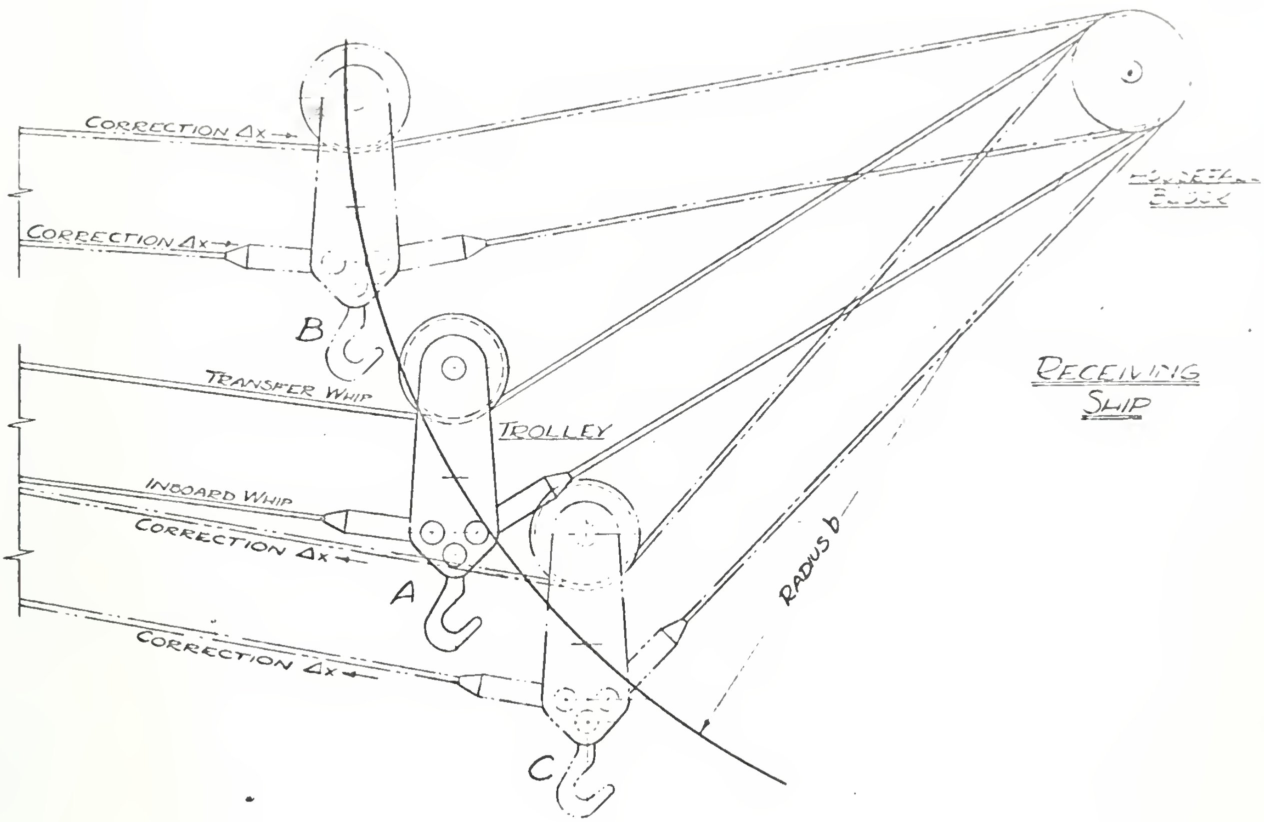

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FIGURE A-5 PHASED TO RECEIVING SHIP
RECEIVING SHIP END

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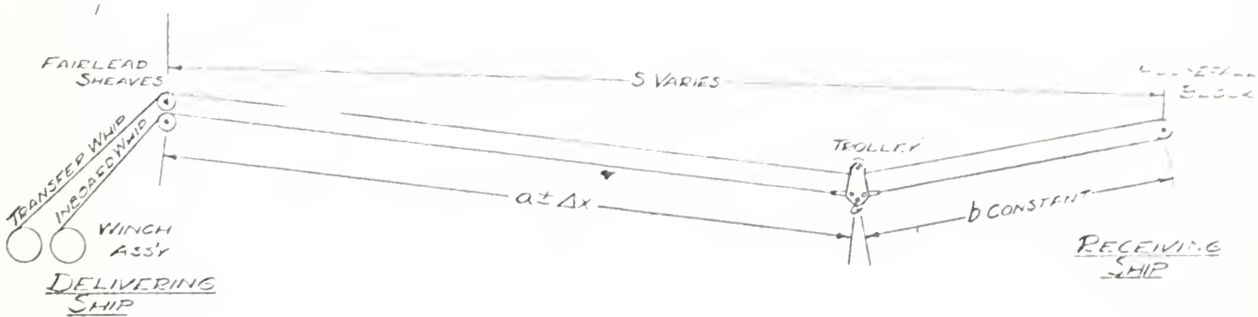
decreased thereby decreasing tension. The trolley moves down along the arc at radius "b" centered at the pin of the housefall block aboard the Receiving Ship. Two lengths "Δx" are taken from the rig: "Δx" from the Transfer Whip and "Δx" from the Inboard Whip. The desired rig tension is restored and the trolley is returned to location "A" without the distance "b" being altered.

Figure A-6 illustrates the effects of tension corrections to the lengths "a" and "b" when phased to the Receiving Ship.



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HD. ENGR.			FIGURE A-6 PHASED TO RECEIVING SHIP, ENTIRE RIG	
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APPENDIX C

PRECEPT: TENSIONED MODIFIED HOUSEFALL MODEL MARK II

(Appendix C of SEVENTH REPORT,
TOTAL LOAD CONTROL, ULTIMATE TRANSFER
RIG . . . June 1963 by author)

C-1 Objectives.

Section 4 of reference (h) contains a development program for the Ultimate Transfer Rig. Model Mark II corresponds to "Model 2", sub-section 4.212. The purposes cited in this past report are basically the same at this date. With only slight modification and augmentation, they are:

C-11 To progress further toward the development of the Ultimate Transfer Rig by providing additional familarization with the requirements to be encountered in the design and development of sizeable operational installations at a larger scale than previous models permitted.

C-12 To further demonstrate the Ultimate Transfer Rig.

C-13 To meetor approach the following potential operational capabilities:

C-131 Installation aboard Service and Combatant Ships as an improved underway replenishment method replacing the conventional Wire and Manila Highline.

C-132 Improved method of rigging and unrigging for the Tensioned Highline method of underway replenishment pending the installation of the large size Tensioned Modified Housefall as the Ultimate Transfer Rig.

C-133 Tensioned Outhaul/Inhaul system for the Tensioned Highline method of underway replenishment pending the installation of the large size Tensioned Modified Housefall as the Ultimate Transfer Rig.

C-134 Improved method of rigging and un-rigging for underway fueling including probe engaging and dis-engaging.

C-135 Improved method of underway fueling hose suspension replacing the Spanwire and Close-in rigs for fueling at sea.

C-136 After extensive testing and proven capability and reliability: Improved method of transferring personnel at sea.

C-14 To provide an installation for test and evaluation toward operational acceptance of the Ultimate Transfer Rig.

C-2 Initial Concept.

Section 2 of reference (d) establishes the initial potential advance of Model Mark II. This basically is the elimination of "the mechanical transmission components between the electric prime movers and the winch drums".

C-3 Major Components.

C-31 Hydraulic Components. Four Vickers PVB 10 variable axial piston pumps will be utilized in the drive transmission, two each for tensioning and transfer flow generation. Two Washington Scientific Industries, Incorporated Series 900 fixed rotary vane motors will be utilized, also, in the drive transmission, one directly driving each winch drum. The ratings of these components are:

Vickers PVB-10 Variable Pump

Theoretical Displacement	1.29 cu in/rev
Flow	To 20 gpm; 10 gpm at rated speed
Operating Pressure	To 1500 psi
Rated Operating Pressure	1000 psi
Operating Speed	To 3600 rpm
Rated Operating Speed	1800 rpm
Weight (dry)	Approx. 29 pounds

WSI Series 900 Motor

Maximum Speed	1000 rpm
Maximum Pressure	2000 psi
Theoretical Volume	25.4 cu in/rev 110 gpm at Max rpm
Theoretical Torque	404.3 in lbs at 100 psi 8086 in lbs at max pres 673.8 ft lbs at max pres
Horsepower	128.3 at max rpm and max pres

A Vickers PVB 5 pressure compensated variable axial piston pump will be utilized as a servo control source for both the tensioning and transfer drive flow generating sub-systems. The ratings of this pump are:

Theoretical Maximum Displacement	.663 cu in/rev
Flow	To 10 gpm (5 gpm at rated speed)
Operating Pressure	To 3000 psi
Rated Operating Pressure	1500 psi
Pressure Compensated Range	250 to 3000 psi
Operating Speed	To 3600 rpm
Rated Operating Speed	1800 rpm

A Vickers VV3-5C-10 pressure compensated variable vane pump will be utilized as a replenishing source for the drive transmission. The ratings of this pump are:

Maximum Operating Pressure - - - - - 1000 psi
Pressure Compensated Range - - - - - 200 to 1000 psi
Maximum Recommended Drive
Speed (Approx.) - - - - - - - - - - - - - 1800 rpm
Minimum Recommended Drive Speed - 600 rpm
Maximum Pump Delivery - - - - - - - 5 gpm

C-32 Electric Components. A 15 HP double ended motor will be utilized to directly drive the variable pumps generating tensioning drive flow. The name plate data for this motor is:

Marathon Electric Manufacturing Corporation
Model No. 284U-TFS-330-BB
Type TFS Frame 284U HP 15 RPM 1760
Cyc 60 PH 3 Volts 208-220/440 WDG H-28441B-3
Duty Cont. Rise 55°C Encl TEFC Brg Ball; Double Shaft

Two 5 HP double ended motors will also be utilized. One to drive the variable pumps generating transfer drive flow; the other to drive the servo control and the replenishing pumps. The name plate data for these motors is:

Marathon Electric Manufacturing Corporation
Model No. 215-TFR-31-BB
Type TFS Frame 215 HP 5 RPM 1740
Cyc 60 PH 3 Volts 208-220/440
Duty Cont. Rise 55°C Encl TEFC Brg Ball; Double Shaft

C-4 Anticipated Capabilities and Design Evolution.

C-41 Wire. Initial operating capabilities and design evolutions are formulated upon the use of 1/8 inch 7 x 19 Strand Core CRES wire rope. The use of 1/8 inch wire rope provides the maximum available theoretical capabilities commensurate with the utilization of readily available facilities and rigging fixtures. With the use of this limited size wire a more flexible rig is established, higher whip speeds are available, and a lower hydraulic line pressure may be utilized than if wire size were increased. It is intended that the design of the winch assembly will permit the exchange of winch drums for a larger wire size if later desired.

C-42 Safety Factor. Relief valve operation is based upon a 5:1 safety factor relative to the ultimate tensile strength of 1/8 inch 7 x 19 Strand Core CRES wire rope.

(1) Ultimate Breaking Strength
1/8 inch 7 x 19 Strand Core CRES wire rope — 1760 lbs

(2) Relieving Tension per rig part
$\frac{(1)}{\text{safety factor 5}}$ — 352 lbs

(3) Relieving Tension total - 2 (parts)(2) - — 704 lbs

(4) Assume nominal operating tension, total - — 660 lbs

(5) Nominal operating tension per rig part - $\frac{(4)}{2}$ — 330 lbs

(6) Nominal operating pressure across hydraulic motor corresponding to (5) - — 800 psi

(7) Torque, each motor, at 800 psi - $\frac{(6)(404.3 \text{ in lb}/100 \text{ psi}}{100 \text{ psi}}$
8 (404.3 in lb) — 3234.4 in lbs

(8) Mean working wire layer radius - $\frac{(7)}{(5)}$ — 9.80 in

(9) Wire per coil at mean working wire layer radius - $2\pi(8)$ — 61.61 in/rev MF or 5.134 ft/rev MF

(10) Variable pump revolutions at full displacement required to drive fixed motor one revolution -

$$\frac{\text{Motor displacement, cu in}}{\text{(Pump displacement, cu in)(pump volumetric efficiency)} \times \text{(Motor volumetric efficiency)}}$$

$\frac{25.4}{(1.29)(.98^*)(.95^*)}$ - — 21.15 revs PV/rev MF

* Interpolated effeciencies from manufacturers' data.

(11) Fixed motor speed corresponding to full flow from one variable pump - $\frac{\text{electric motor speed, rpm}}{(11)}$ -
$\frac{1760}{21.15}$ - — 83.21 rpm

(12) Whip speed corresponding to full flow from one variable pump - (9)(11) - (5.134 ft/rev) x (83.21 rpm) - — 427.20fpm

(13) Maximum fixed motor speed to be encountered due to variable pump generated hydraulic flow: Condition: Transfer Whip only, phased to Delivering Ship, transfer at maximum speed driving in the same direction as the simultaneous maximum tensioning flow. Three variable pumps at maximum output are driving one fixed motor.
3(11) = 3(83.21 rpm) - — 249.63 rpm

(14) Transfer whip speed corresponding to (13).
$3(12) = 3(427.20 \text{ fpm}) =$ 1281.60 fpm

(15) Maximum tensioning drive power input -

$$\frac{\text{Pres x Flow, cu in/min}}{\text{efficiency, pump x efficiency, motor x } 396,000}$$

$$\frac{(1000 \text{ psi}^{**})(2 \text{ pumps})(1.29 \text{ cu in/rev per pump})(1760 \text{ rpm})^{*}}{(.85^{*})(.88^{*})(396,000)}$$ 15.32 HP

(16) Maximum transfer drive power input - Condition: Transfer drive system is operating at maximum speed, one variable pump, (paying in) is working against a pressure of 1000 psi, and that rig loading overhauls the paying out winch drum.

$$\frac{(1000 \text{ psi})(1.29 \text{ cu in/rev})(1760 \text{ rpm})}{(.85)(.88)(396,000)} -$$

$$\frac{(\text{tension/part, lbs})(\text{whip spd, fpm})(\text{effic. motor})(\text{effic. pump})}{33000 \text{ ft lbs/min per HP}}$$

$$7.66 - \frac{(330 \text{ lbs})(427.20 \text{ fpm})(.88)(.85)}{33000 \text{ ft lbs/min per HP}}$$

$7.66 - 3.20 =$ 4.46 HP

(17) Relief valve setting - $\frac{(2)(6)}{(5)}$ + Replen pressure

$\frac{(352 \text{ lbs})(800 \text{ psi})}{(330 \text{ lbs})} + (200 \text{ psi}) =$ 1053.3 psi

(18) Winch drums are based upon capability of operating with a separation of 200 feet. To insure availability of wire with load catenary, winch drum capacity is based upon a separation of 250 feet. A separation of 100 feet will be assumed to be the normal operating separation for determining the design mean wire layer radius, (9.80 inches (8) above) to be utilized upon both winch drums with the trolley at the Receiving Ship. With the trolley at the Delivering Ship, the Transfer Whip must extend twice the capacity separation of 250 feet. To accomodate this requirement the Transfer Whip winch drum must have a minimum capacity of 500 feet.

As the Inboard Whip extends across the ship separation a maximum of one part only, when the trolley is at the Receiving Ship, the Inboard Whip winch drum must have a minimum capacity of 250 feet.

* Interpolated efficiencies from manufacturers' data.
** 800 psi operating pressure plus 200 psi replenishing pressure.

Considering the above, winch drum wire loadings under several conditions are:

Condition:	Wire upon winch drums Transfer Whip	Inboard Whip
A. Stowed aboard Delivering Ship	500 ft	250ft
B. Rigged, 250 ft separation: trolley at Delivering Ship	0	250 ft
C. Rigged, 250 ft separation: trolley at Receiving Ship	250 ft	0
D. Rigged, 100 ft separation: trolley at Receiving Ship	400 ft	150 ft

It is desired that the Transfer Whip and Inboard Whip winch drums have 400 ft and 150 ft (condition D) upon them with operating wire layer radii of 9.80 inches (8) above.

Through use of the formula (reference (f)):

$L = (A - B) \times A \times C \times F$
L = capacity of drum in feet
A = Depth of wire spooled on drum
B = Barrel diameter
C = Drum width
D = Wire diameter
F = Spooling factor $= \frac{.262}{D^2}$ (controlled winding)

with the stipulations specified above, the following winch drum dimensions are desired:

	Transfer Whip	Inboard Whip
B:	18.73 inches (18-3/4 inches)	19.23 inches (19-1/4 inches)
C:	3 inches	3 inches

Further use of the above formula provides the following table of winch drum capacities by layers:

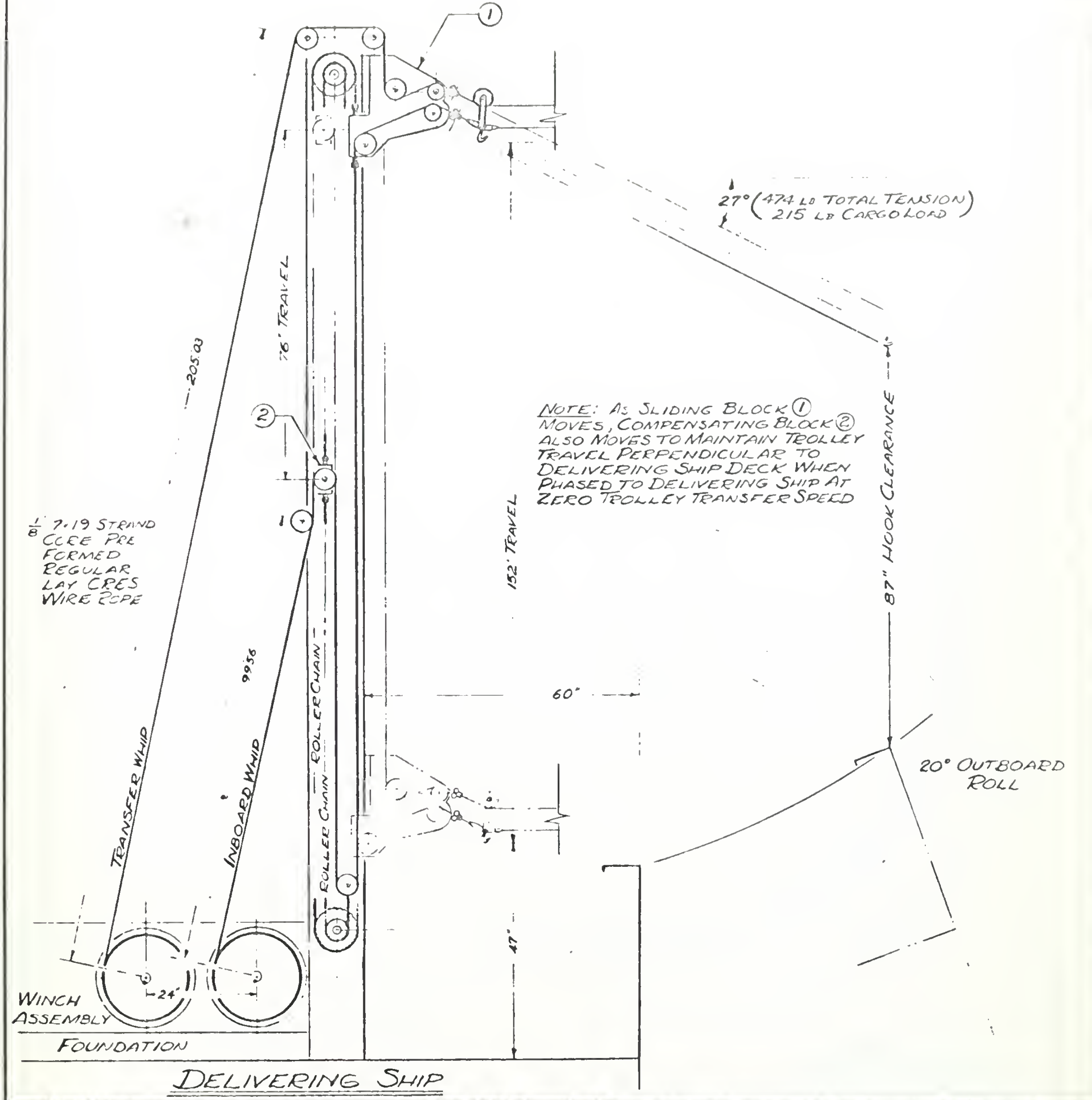
Com-plete wire layers	Transfer Whip feet	pitch dia, in	feet/ coil	Inboard Whip feet	pitch dia, in	feet/ coil
1	118.54	18.85	4.94	121.69	19.35	5.07
2	238.65	19.10	5.00	244.94	19.60	5.13
3	360.34	19.35	5.07	369.77	19.85	5.20
4	483.60	19.60	5.13	496.17	20.10	5.26
5	608.43	19.85	5.20	624.15	20.35	5.33
6	734.83	20.10	5.26	753.69	20.60	5.40
7		20.35	5.33		20.85	5.46

Ideally the Transfer Whip winch drum should be loaded with two more layers than the Inboard Whip winch drum when the trolley is at the Receiving Ship; or when the rig is stowed with the trolley at the Delivering Ship and at zero separation.

With the rig stowed aboard the Delivering Ship, trolley at the Delivering Ship, zero separation with fairleads rigged an ideal condition would be to have four complete layers of wire plus eight coils stowed upon the Transfer Whip winch drum and two complete layers plus eight coils stowed upon the Inboard Whip winch drum. With a separation of from 41.60 feet (8 coils x 5.20 ft/coil) to 164.85 feet (41.60 plus the applicable layer of either winch drum) both winch drums will be operating at the design mean layer radius (9.80in.) when the trolley is at the Receiving Ship.

The winch drums are to have an overall diameter of 21-1/2 inches. For maintenance purposes, the trolley can be removed from the rig, the whip bitter ends temporarily connected together, and the entire length of wire required can be stowed upon either of the winch drums with ample flange clearance.

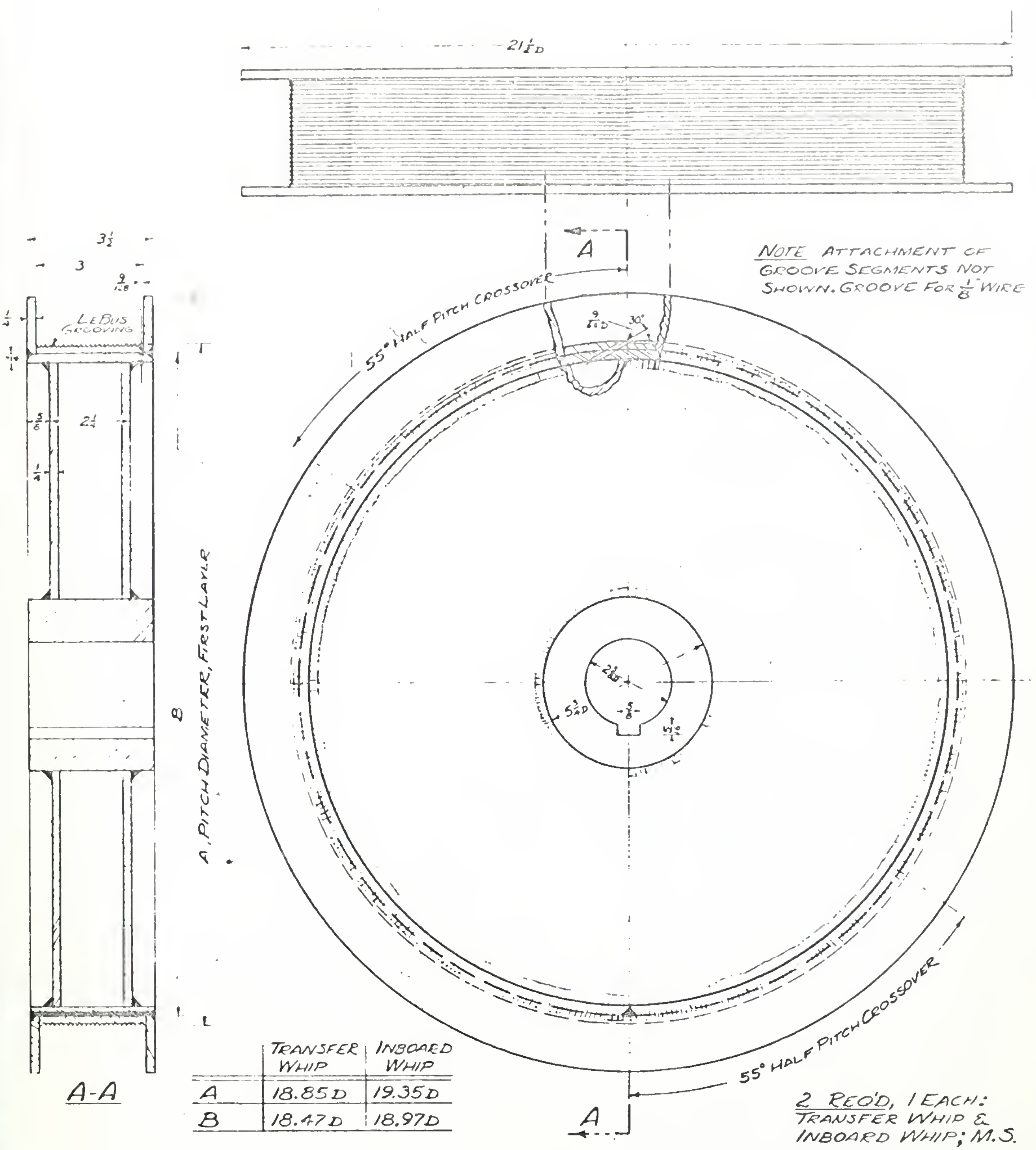
Figures C-1 through C-4 illustrate currently contemplated arrangements and components of the Tensioned Modified Housefall Model Mark II.



REV. / DESCRIPTION / DATE / APP'D — REVISIONS	APPROVED — DATE 5-27-63 — DESIGN SUPT.	SAN FRANCISCO NAVAL SHIPYARD SAN FRANCISCO 24, CALIFORNIA	
	OFFICIAL — SIGNATURE HD. ENGR. BR. SUP. SEC. SUP. REVIEWED DRAWN	TENSIONED MODIFIED HOUSEFALL MODEL MARK II CONTEMPLATED DELIVERING SHIP RIG ARRANGEMENT	
REPLENISHMENT-AT-SEA CONTRACT GUIDANCE DRAWING		BUREAU OF SHIPS NO. 254 9200 100002 SCALE — SHEET — OF	REV.

72

FIGURE C-1



	Transfer Whip	Inboard Whip
A	18.85D	19.35D
B	18.47D	18.97D

REV.	DESCRIPTION	DATE	APP'D
	REVISIONS		

REPLENISHMENT-AT-SEA
CONTRACT GUIDANCE DRAWING

APPROVED	DATE 5-28-63
	DESIGN SUPT.
OFFICIAL	SIGNATURE
HD. ENGR.	
BR. SUP.	
SEC. SUP.	
REVIEWED	
DRAWN	

SAN FRANCISCO NAVAL SHIPYARD
SAN FRANCISCO 24, CALIFORNIA

TENSIONED MODIFIED HOUSEFALL MODEL MARK II WINCH DRUMS

BUREAU OF SHIPS NO.			REV.
254	9200	100003	

SCALE | SHEET 1 OF 1

FIGURE C-2

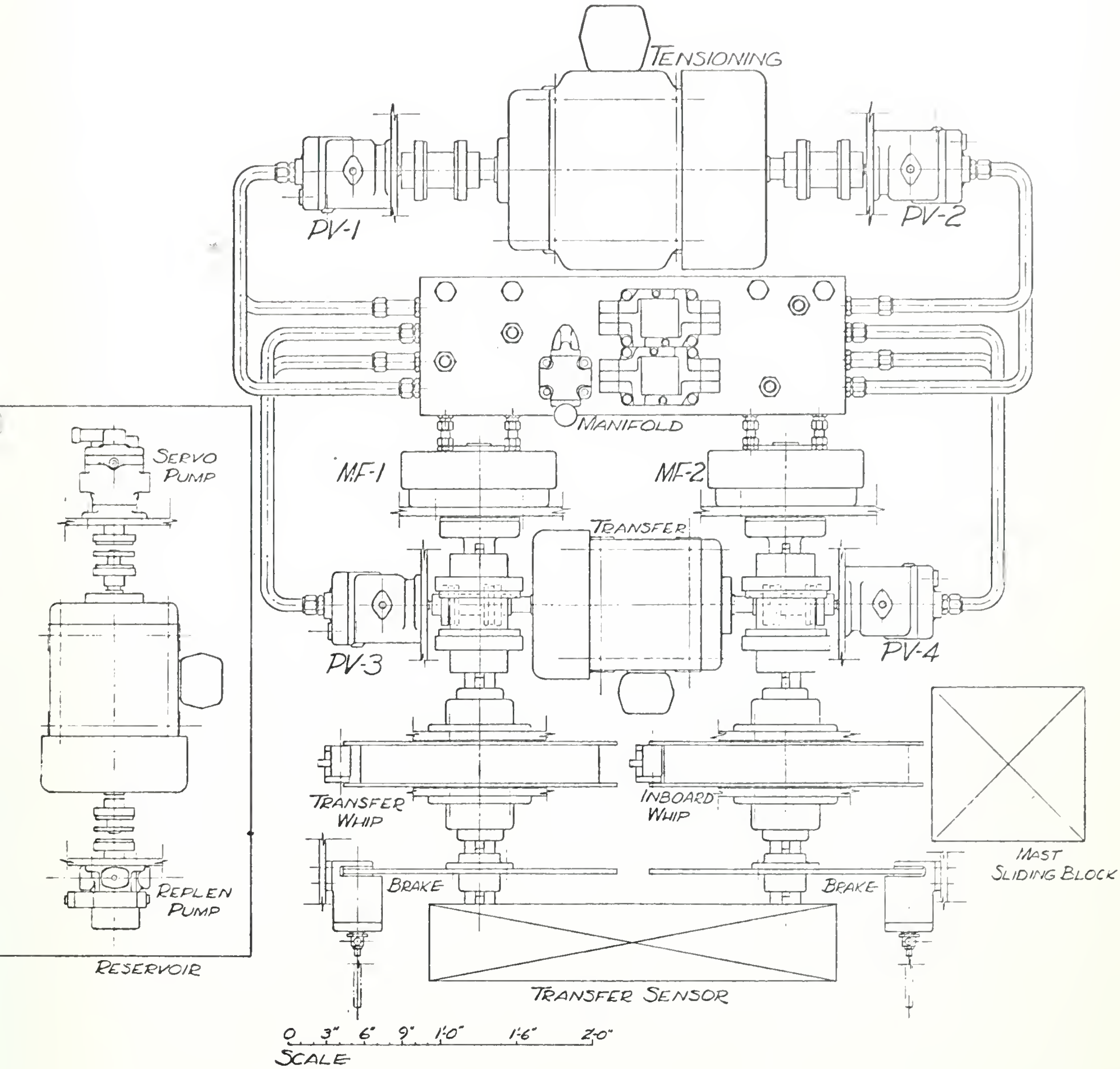


FIGURE C-3
TENSIONED MODIFIED HOUSEFALL
MODEL MARK II
CONTEMPLATED MACHINERY ARRANGEMENT

APPENDIX D

TOTAL LOAD CONTROL SYSTEM

(EIGHTH REPORT, TOTAL
LOAD CONTROL . . . prepared December
1963 by author).

SHILLINGER TOTAL LOAD CONTROL SYSTEM

EIGHTH AND FINAL REPORT

March 1964

ABSTRACT

This report has been prepared to comply with San Francisco Naval Shipyard contract SFNS-3280-M34-21-X. Basically this contract requires a drawing presenting the Electrical Schematic Wiring Diagram of the Shillinger Tensioned Modified Housefall Model Mark II along with a detailed word description of the principles and methods of operation of this model.

The main drawing presenting the Electrical Schematic Wiring Diagram is figure 2 of this report. The other drawings combine orientation, electrical, hydraulic and mechanical control features of this model. These additional drawings are essential for an explanation of the operation of this model.

This report supplements Appendix C of reference (a) which contains the initial description of the contemplated Mark II model.

Section 7 of this report summarizes the status of this project at this time.

No attempt is made in this report to describe "phasing" which, in essence, is the prime justification for this project with its capability of transferring delicate cargo, with rig simplicity and expediency, from ship to ship at sea under adverse conditions. Appendix A of reference (b) explains this quality.

REFERENCES

(a) Seventh Report - Total Load Control - Ultimate Transfer Rig - Buships Task 2700 - Contract NObs 88009 - Navy Case 34129 by Lt. G. L. Shillinger, Jr., June 1963.

(b) Sixth Report - Ultimate Transfer Rig - Navy Case 34129 - Buships Task 2700 - Contract NObs 88009 by Lt. G. L. Shillinger, Jr., March 1963.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

NOMENCLATURE

Acc	Accumulator
BR-	Brake
CA-	Control Actuator
CL-	Clutch
CV-	Check Valve
DA-	Differential Actuator
D/S	Delivering Ship
DSV-	Double Solenoid 3-position 4-way Valve; Spring Loaded to center position.
EM-	Electric Motor
F	Filter
FS-	Flexible Shaft
HYD	Hydraulic
H2V-	Hydraulic Pilot Operated 2-way Valve
H3V-	Hydraulic Pilot Operated 3-way Valve
MF-	Hydraulic Motor, Fixed Displacement
MTEL	Manual Tensioning Control Lever
MTRL	Manual Transfer Control Lever
M4V-	Manually Operated 4-way Valve
NC	Normally Closed
NO	Normally Open
OS-	Operator Switch
P-	Pinion, Drives Spiroid Gear; 50:1 speed reduction
PHV-	Phase Shift Double Hydraulic Pilot Operated 3-way Valve
PS-	Programming Switch; Combination adjustable snap acting switch and rotary cam assembly.
PV-	Hydraulic Pump, Variable Displacement
R-	Relay
RC-	Relay operated contacts; RC31 and RC32 are operated by relay R3, etc.
RE-	Restriction
RV-	Relief Valve
R/S	Receiving Ship
SA-	Snap Acting Switch
SG-	Spiroid Gear
SOL-	Solenoid
SV-	Servo Valve, Mechanical
SS3V-	Single Solenoid 3-way Valve
SS4V-	Single Solenoid 4-way Valve
TX-	Synchro Transmitter

. GENERAL DESCRIPTION

1.1 Description (Figure 1)

The Tensioned Modified Housefall Model Mark III Winch Assembly has two winch drums: Transfer Whip and Inboard Whip. Each of these winch drums is driven directly by a fixed displacement low speed high torque hydraulic motor (MF1 and MF2); MF1 drives the Transfer Whip Winch Drum; MF2 drives the Inboard Whip Winch Drum. Hydraulic Flows to and from MF1 and MF2 are through the main manifold which combines drive hydraulic flows for tensioning with phasing and transfer.

The main manifold also provides for the mounting of 11 valves:

2 Phase Shift Valves (PHV1 and PHV2)
4 Relief Valves (RV1 through 4)
4 Check Valves (CV1 through 4)
1 Manually Operated 4-Way Valve (M4V1)

At the Main Manifold, bosses are provided for the following accessories:

Tension sensing (from Pay-in Pressure side of MF1 to Automatic Tensioning Control System).

Bleeding
Pressure Gage Connections
Pressure Transducer Connections

1.2 Tensioning Flow Generation

The Tensioning Flow Generator consists of an electric motor (EM1) and two hydraulic variable pumps (PV1 and PV2). The two pumps are driven at exactly the same speed as they are coupled directly to a common shaft. The stroking mechanisms for these pumps are also mechanically attached and synchronized such that the flow from PV1 is always theoretically equal to the flow from PV2. The flows from the two pumps are in the same direction to the main manifold.

The flow from PV1 always drives MF1. The flow from PV2 drives either MF1 <u>or</u> MF2 depending upon phasing:

1.21 Phased to Delivering Ship

Valves PHV1 and PHV2 are in position A. Both pumps (PV1 and PV2) drive fixed hydraulic motor MF1.

1.22 Phased to Receiving Ship

Valves PHV1 and PHV2 are in position B. Pump PV1 drives fixed hydraulic motor MF1. Pump PV2 drives fixed hydraulic motor MF2.

1.3 Transfer Flow Generation

The Transfer Flow Generator consists of an electric motor (E[illegible]) and two hydraulic variable pumps (PV3 and PV4). These pumps are coupled to a common shaft and their stroking mechanisms are mechanically linked and synchronized such that the flow from PV3 is always theoretically equal to the flow from PV4. However the flows from PV3 and PV4 are in opposite directions to the main manifold. Simultaneous flows from PV3 and PV4 theoretically drive fixed hydraulic motors MF1 and MF2 at the same speed but in opposite directions. PV3 drives MF1 while PV4 drives MF2.

1.4 Final Drive

The flows from the tensioning flow generator and the transfer flow generator are integrated at the main manifold and properly distributed to MF1 and MF2. Tensioning correction with phasing and transfer occur simultaneously.

1.5 Auxiliaries

Electric Motor EM3 drives pressure compensated variable pumps (PV5 and PV6) to provide Servo Control (PV5) and Hydraulic Replenishment (PV6) sources.

1.51 Servo Pump

PV5 provides source pressure for phase shifting, operation of the tensioning control system, operation of the transfer control system and operation of the sliding block. Detailed descriptions of the operations of these systems will appear in subsequent paragraphs.

1.52 Replenishment Pump

PV6 provides make-up flow to the main drive transmission to maintain hydraulic tightness. Replenishment flow is from PV6 through CV(S)1,2,3 and/or 4 to the main drive hydraulic lines. Flow from PV6 is also used to disengage spring set brakes as further described below.

1.53 Brakes

Spring set brakes are provided for each of the winch drums and for the sliding block drive:

BR1 Transfer Whip Winch Drum
BR2 Inboard Whip Winch Drum
BR3 Sliding Block Drive

Manually operated levers are provided for releasing brakes following a hydraulic failure.

2. FAIL SAFE (Figures 1 through [illegible])

Fail safe features include the following:

2.1 <u>Spring Set Brakes</u>: The brakes BR1, BR2 and BR3 are set by spring force and are hydraulically released by replenishing system pressure. Hand release levers are also provided so that the brakes may be manually released in the event that they lock due to a hydraulic failure.

For the entire duration of the Replenishment-at-Sea operation, the winch drum brakes BR1 and BR2 are normally held disengaged by hydraulic pressure.

To avoid parting wires following a hydraulic failure, brakes BR1 and BR2 should be designed to slip with a sufficient safety factor to prohibit increasing tension from building to wire breaking strength.

The brakes are also dependent upon the availability of hydraulic servo pressure for their operation. The spool of valve H3V1 is spring set to position A which isolates and drains the brakes. With servo pressure available the spool of valve H3V1 is pilot shifted to position B thereby permitting replenishment pressure to release the winch drum brakes, BR1 and BR2, and providing an operating pressure source for the sliding block brake, BR3.

2.2 <u>Control Linkages spring loaded to "stop"</u>: The control linkages stroking the tensioning variable pumps PV1 and PV2, and the transfer variable pumps PV3 and PV4, are spring loaded to their center or "stop" positions. The A and B ends of the tensioning and transfer control actuators CA1 and CA2 are connected through valves H2V1 and H2V2 respectively. The spools of valves H2V1 and H2V2 are spring loaded to positions A which permits free flow of hydraulic fluid from one end of the control actuator, CA1 and CA2, to the other end of the same control actuator. When servo pressure is available the spools of valves H2V1 and H2V2 are pilot shifted by servo pressure from spring loaded positions A to positions B thereby blocking the hydraulic short circuit between the ends of each control actuator.

In the event of loss of hydraulic servo pressure the spools of valves H2V1 and H2V2 spring from operating positions B to spring loaded positions A thereby short circuiting the control actuators CA1 and CA2. The spring loading forces the control linkages to their center or "stop" positions thus stopping the hydraulic flows to and from the hydraulic motors, MF1 and MF2. The winch drums are stopped.

2.3 Electric control power essential. Electric control power is essential for the control systems to operate as these systems include solenoid operated valves. Valve SS3V1 isolates the control systems from the servo pressure source in the event of failure of electric power to the control systems. The spool of valve SS3V1 is spring loaded to position A thereby isolating and draining the hydraulic lines of the control systems. When electric control power is available solenoid SOL1, electric circuit branch D1, is energized shifting the spool of valve SS3V1 from spring loaded position A to position B thus permitting servo hydraulic flow to the control systems.

In the following descriptions concerning the electric hydraulic sequences for phase shifting, tensioning, stop correcting, transfer control and sliding block systems; the fail safe features described above will not be referred to. Although the fail safe features are important they do not contribute to the operation of the systems described below beyond the extent already stated above. Thus the descriptions of electric hydraulic sequences which follow are dependent upon the existence of adequate control electric and hydraulic servo and replenishing supply sources.

3. PHASE SHIFT (Figures 1 through 4)

The phase shift occurs during both manual and automatic operation of the transfer system. The phase shift occurs automatically at a fixed position of the trolley with respect to the delivering ship. The point at which the phase shift occurs is dictated by the transfer sensor. Programming switch PS1 opens or closes to initiate the phase shift. As the trolley moves away from the delivering ship the inboard whip is paid out from its winch drum. As the inboard whip winch drum rotates it also rotates flexible shaft FS1 which directly drives pinion P1 further driving spiroid gear SG1 rotating the cam of programming switch PS1. The speed reduction through the pinion and spiroid gear is 50:1, thus one rotation of the cam of programming switch PS1 corresponds to less than 250 feet of travel of the trolley from the delivering ship. The cam and switch of programming switch PS1 are set to operate as the trolley passes through Point I, a fixed distance from the delivering ship.

Considering the three dimensional aspect this point is any point on a spherical surface with the center for this spherical surface being the upper tangent of the inboard whip fairlead sheave.

The sequence of operations associated with the phase shift are:

With the trolley enroute from its initial position at the delivering ship (zero distance from the delivering ship):

In the vicinity of the delivering ship, during the initial transfer of the trolley toward the receiving ship, tensioning is phased to the delivering ship.

PS1 is open, electric circuit branch D2 is broken and solenoid SOL2 is not energized: valve SS4V1 is in its spring loaded position A. Pilot pressure through valve SS4V1 is to the A end of valves PHV1 and PHV2. The pilot line to the B end of valves PHV1 and PHV2 is drained through valve SS4V1, with valves PHV1 and PHV2 in position A. Thus the hydraulic flow generated by tensioning variable pump PV2 serves fixed hydraulic motor MF1 which drives the transfer-whip winch drum. With PS1 open both tensioning variable pumps, PV1 and PV2, drive the transfer whip winch drum and tensioning corrections are made without affecting the position of the trolley relative to the delivering ship.

When the trolley reaches point I (as the trolley moves away from the delivering ship) programming switch PS1 closes completing the electric circuit branch D2 thereby energizing SOL2: valve SS4V1 is shifted from position A to B. Through valve SS4V1 pilot pressure now flows to ends B of valves PHV1 and PHV2 while ends A are drained. Valves PHV1 and PHV2 shift from position A to position B. Tensioning variable pump PV2 now serves fixed hydraulic motor MF2 which drives the inboard whip winch drum. With PS1 closed each of the tensioning variable pumps drives a winch drum and tensioning corrections are made without affecting the position of the trolley relative to the receiving ship.

As the trolley passes through point I on its return towards the delivering ship, programming switch PS1 is opened de-energizing solenoid SOL2. Valve SS4V1 is shifted from position B to spring loaded position A. In a manner similar to that described above but opposite, phasing is thus shifted from the receiving ship to the delivering ship as the trolley approaches the delivering ship.

4. TENSIONING (Figures 1 through 5)

4.1 Manual Operation

4.11 Place operator switch OS1 in "Manual" (open) position. Electric circuit branch D3 is broken, therefore solenoids SOL3, SOL4 and SOL5 are not energized. Valves SS3V2, SS3V3 and SS3V4 have their spools in spring loaded positions A. With the spool of valve SS3V2 in position A return hydraulic flow is from servo valve SV1 while return flow from servo valve SV2 is blocked. Valves SS3V3 and SS3V4 permit flow through servo valve SV1 to and from the tensioning control actuator CA1 while isolating servo valve SV2.

4.12 To "pay out": Manually move manual tensioning control lever MTEL in "pay out" direction. Through a mechanical linkage with suitable reduction the spool of servo valve SV1 is also moved toward direction A establishing hydraulic

flow through servo valve SV1 and SS3V3 to end A of tensioning control actuator CA1. The B end of the tensioning control actuator CA1 drains through valve SS3V4, servo valve SV1 and valve SS3V2 to the reservoir. The rod of the tensioning control actuator CA1 moves in the "pay out" direction simultaneously stroking tensioning variable pumps PV1 and PV2. The sleeve of servo valve SV1 is also positioned by movement of the rod of the tensioning control actuator CA1. As the rod of the tensioning control actuator CA1 approaches the position dictated by operator displacement of the manual tensioning control lever MTEL, the hydraulic flow to and from the tensioning control actuator CA1 is nullified. The final response position of the control actuator CA1 rod, the tensioning drive hydraulic flow and the resultant payout speed of fixed hydraulic motor MF1 if phased to the delivering ship or of fixed hydraulic motors MF1 and MF2 if phased to the receiving ship, correspond to the operator displacement of the manual tensioning control lever MTEL.

4.13 To "<u>pay in</u>": Manually move manual tensioning control lever MTEL in "pay in" direction. The spool of servo valve SV1 is also moved toward direction B. Hydraulic flow through servo valve SV1 and valve SS3V4 is to the B end of the tensioning control actuator CA1. The A end of the tensioning control actuator CA1 is drained through valve SS3V3, servo valve SV1 and valve SS3V2 to the reservoir. The rod of the tensioning control actuator CA1 moves in the "pay in" direction simultaneously stroking tensioning variable pumps PV1 and PV2. The sleeve of servo valve SV1 is also positioned by movement of the rod of the tensioning control actuator CA1. As the rod of the tensioning control actuator CA1 approaches the position dictated by operator displacement of the manual tensioning control lever MTEL, the hydraulic flow to and from the tensioning control actuator CA1 is nullified. The final response position of the control actuator CA1 rod and the resultant pay in speed correspond to the operator displacement of the manual tensioning control lever MTEL.

4.2 <u>Operation of Inboard Whip Winch Drum Alone</u>

The main hydraulic drive transmission does not provide any means of operating the Inboard Whip winch drum only. For limited purposes, such as placing wire upon or removing wire from the Inboard Whip winch drum, a means of driving fixed hydraulic motor MF2 alone at limited capacity is provided.

4.21 To operate Inboard Whip winch drum alone, manually shift Transfer Whip bypass valve M4V1 from position A to position B. Through manipulation of the manual transfer control lever MTRL the Inboard Whip winch drum may be rotated alone at limited speeds.

With the shift of the Transfer Whip bypass valve M4V1 from position A to position B, flow from the replenishing system through valve M4V1 shifts the spool of valve H3V2 from its spring loaded position A to position B by pilot operation. The Transfer Whip winch drum brake BR1 is then drained through valve H3V2 to the reservoir. The Transfer Whip winch drum brake BR1 is then applied by spring action. Simultaneously, through the Transfer Whip bypass valve M4V1, the pay-in and pay-out sides of fixed hydraulic motor MF1 are cross connected. Thus with valve M4V1 in position B fixed hydraulic motor MF1 is short circuited and the spring set brake BR1 is applied to the Transfer Whip winch drum. With manipulation of the manual transfer control lever MTRL only fixed hydraulic motor MF2, driving the Inboard Whip winch drum, will be operated.

4.22 To restore to normal operating conditions, manually shift Transfer Whip bypass valve M4V1 from position B to position A. The pilot line to valve H3V2 is now isolated from the replenishing pressure by the spool of valve M4V1. Thus the pilot to valve H3V2 drains and valve H3V2 shifts from position B to its spring loaded position A. Flow from the replenishing supply now hydraulically releases the Transfer Whip winch drum brake BR1. With the spool of the Transfer Whip bypass valve M4V1 in position A, the cross connection between pay-in and pay-out sides of fixed hydraulic motor MF1 is blocked. Thus the winch assembly is now restored for normal Tensioned Modified Housefall operation.

4.3 Automatic Operation

4.31 Place operator switch OS1 in "Automatic" (closed) position, electric circuit branch D3 is completed. Solenoids SOL3, SOL4 and SOL5 are energized. The spools of valves SS3V2, SS3V3 and SS3V4 are shifted from their spring loaded positions A to positions B. The spool of valve SS3V2 now permits servo valve SV2 to drain while the return flow from servo valve SV1 is blocked. The spools of valves SS3V3 and SS3V4 permit flow through servo valve SV2 to and from the tensioning control actuator CA1 while isolating servo valve SV1.

4.32 If tension is at desired value: The spool of servo valve SV2 is located at its central position producing balanced flow to both ends of the tensioning control actuator CA1 through valves SS3V3 and SS3V4; theoretically zero hydraulic flow to both ends of the tensioning control actuator CA1. The strokes of tensioning variable pumps PV1 and PV2 are at the zero flow center position. The spool of servo valve SV2 is positioned by spring force opposed by the hydraulic force produced by the pressure of the pay in side of the fixed hydraulic motor MF1 at differential actuator DA1.

4.33 As disturbances increase rig tension: Pressure in pay in side of fixed hydraulic motor MF1 increases causing the piston rod of differential actuator DA1 to move toward direction A. The spool of servo valve SV2 is moved in direction A by the greater force of the piston of differential actuator DA1 against spring compression, due to the increased pressure at the pay in side of the fixed hydraulic motor MF1. Servo hydraulic flow is established through servo valve SV2 and valve SS3V to end A of the tensioning control actuator CA1. The B end of the tensioning control actuator CA1 drains through valve SS3V4, servo valve SV2 and valve SS3V2 to the reservoir. The rod of the tensioning control actuator CA1 moves in the "pay out" direction simultaneously stroking tensioning variable pumps PV1 and PV2. The sleeve of servo valve SV2 is also positioned by movement of the rod of the tensioning control actuator CA1. As the rod of the tensioning control actuator CA1 approaches the position dictated by the rig tension, the hydraulic flow to and from the tensioning control actuator CA1 is nullified. The final response position of the control actuator CA1 rod and the resultant pay out speed correspond to the quantity of tension discrepancy above the desired value of tension.

4.34 As disturbances decrease rig tension: Pressure in pay in side of fixed hydraulic motor MF1 decreases causing the piston rod of differential actuator DA1 to move toward direction B. The spool of servo valve SV2 is moved in direction B by the greater force of the spring against the piston of differential actuator DA1, due to the decreased pressure at the pay in side of the fixed hydraulic motor MF1. Servo hydraulic flow is established through servo valve SV2 and valve SS4V to end B of the tensioning control actuator CA1. The A end of the tensioning control actuator CA1 drains through valve SS3V3, servo valve SV2 and valve SS3V2 to the reservoir. The rod of the tensioning control actuator CA1 moves in the "pay in" direction simultaneously stroking tensioning variable pumps PV1 and PV2. The sleeve of the servo valve SV2 is also positioned by movement of the rod of the tensioning control actuator CA1. As the rod of the tensioning control actuator CA1 approaches the position dictated by the rig tension, the hydraulic flow to and from the tensioning control actuator CA1 is nullified. The final response position of the control actuator CA1 rod and the resultant pay in speed correspond to the quantity of tension discrepancy below the desired value of tension.

5. TRANSFER (Figures 1 through 4)

5.1 Transfer Sensor

5.11 The transfer sensor receives mechanical inputs from the winch drums. Revolutions of each of the winch drums are transmitted to the transfer sensor by flexible shafts FS1 and FS2. With these inputs the transfer sensor basically determines the movement of the trolley for control and indicator applications. "Position" of the trolley for purposes

of this description is the distance between the trolley and one of the replenishing ships as measured along the rig wires.

The position of the trolley relative to the delivering ship is obtained from the revolutions of the Inboard Whip winch drum.

The position of the trolley relative to the receiving ship is obtained as the difference between the revolutions of the Transfer Whip winch drum and the Inboard Whip winch drum.

Through cam and programming arrangements the Transfer sensor accomplishes the following control evolutions:

For both Manual and Automatic Tensioning and Transfer Drive:

5.111 Performs the initial switch evolution to shift phasing at the proper location of the trolley during its transfer travel.

5.112 Stop correction. With the manual transfer control lever MTRL in the stop (vertical center) position and when in automatic transfer control with switch OS3 in the stop position. The transfer sensor detects unwanted movement of the trolley relative to the particular ship to which the system is phased.

5.113 During automatic transfer control, the transfer sensor dictates the location at which the trolley starts its deceleration preparatory to the landing of the trolley at the ship toward which the trolley is moving.

The control outputs from the transfer sensor are electrical signals produced by the opening or closing of cam operated programming switches PS1 through 5. Additional details of the operation of the transfer sensor will be included in the description of applicable portions of the system which are dependent upon the transfer sensor.

5.12 Remote indication: Synchro transmitters TX1 and TX2 provide synchro signals for remote indication of trolley position. A dial driven by a synchro receiver wired to Synchro transmitter TX1 indicates the distance between the trolley and the delivering ship. A similar arrangement utilizing synchro transmitter TX2 provides an indication of the distance from the trolley to the receiving ship.

5.2 Manual Operation

5.21 Control Operator Inputs

5.211 Operator places switch CS2 in "Manual" position. Solenoids SOL6 and SOL7 are not energized as electric circuit branch D4 is open. Branch FH is open. Valves SS3V5 and SS3V6 thus have their spools in spring set positions A permitting flow to and from servo valve SV3 through valves SS3V5 and SS3V6 to and from the transfer control actuator CA2 while isolating DSV2. Further details of the electrical circuitry involved will follow.

5.212 To "Deliver," (Move trolley toward receiving ship), operator moves manual control lever MTRL in "deliver" direction. Through A mechanical linkage with suitable reduction the spool of servo valve SV3 is also moved toward direction A establishing hydraulic flow through servo valve SV3 and valves SS3V5 to end A of transfer control actuator of CA2. Hydraulic oil from the B end of the transfer control actuator of CA2 flows to the reservoir through valve SS3V6 and servo valve SV3. The rod of control actuator CA2 moves simultaneously stroking transfer variable pumps PV3 and PV4. The sleeve of servo valve SV3 is also positioned by movement of the rod of control actuator CA2 as the rod of transfer control actuator CA2 approaches the dictated position. The flow to and from the transfer control actuator CA2 is nullified. The final response position of the transfer control actuator CA2 rod and the resultant transfer drive flow corresponds to the manual transfer control lever MTRL displacement by operator.

5.213 To "Return," (move trolley toward delivering ship) similar to paragraph 2 above but the manual transfer control lever MTRL is moved in "return" direction. System responds in opposite direction. The final response position of CA2 rod and the resultant transfer drive flow corresponds to the manual transfer control lever MTRL displacement by operator.

5.3 Stop Correction (Figures 2 and 4)

5.31 The purpose of the stop correction feature is to provide the capability of stopping the trolley at any arbitrary intermediate point between the replenishing ships. Ideally this correction should not be required. The inclusion of this feature is to compensate for differences in the volumetric efficiencies among the major hydraulic drive components.

Through the transfer sensor undesired movement of the trolley is detected. Corrective action consists of automatic adjustment of the neural (stop) position of the transfer control linkage.

5.32 Operation

The stop correction system is operable when:

5.321 In Manual Transfer Control Operator switch OS2 is in "manual" position and the manual transfer control lever MTRL is in its "stop" position.

5.322 In Automatic Transfer Control: Operator switch OS2 is in "Auto" position, operator switch OS3 is in "stop" position, and the transfer control linkage is at or near its center position with normally closed snap acting switches SA3 and SA5 held open.

5.33 in Manual Transfer Control, the electrical circuit branch D4 from F to K is completed by OS2 in "manual" position and snap acting switch SA1 being closed. Snap acting switch SA1 is closed when the manual transfer control lever MTRL is at (or in close proximity) of the "Stop" position. From T to ground D.C. current flows through branch T1 with relay operated contacts RC26 in their normally closed position.

5.34 In Automatic Transfer Control, the electrical circuit branch D4 from F to K is completed by OS2 in "Automatic" position with relay operated contacts RC24 in their relay operated closed position and relay operated contacts, RC 51 and RC 61 in their normally closed position.

From T to ground D.C. current flows through branch T2 with relay operated contacts RC 25 in their relay operated closed position and relay operated contacts RC 52 and RC 62 in their normally closed position. D.C. electric current flows from P through electric circuit branch P1 if relay R1 is not energized or through electric circuit branch P2 if relay R1 is energized thence through branch T1 if in "Manual" or branch T2 if in "Automatic" to ground.

5.35 Phasing. Energization of relay R1 is dependent upon phasing.

5.351 If phased to the delivering ship, programming switch PS1 is open and no current flows through electric circuit branch D2. Relay R1 is not energized: relay operated contacts RC11 are closed while relay operated contacts RC12 are open. Hence

electric circuit branch P1 is closed while branch P2 is not. Clutch CL1 is engaged therefore the cams of programming switches PS2 and PS3 are driven through gears by flexible shaft FS1 from the Inboard Whip winch drum.

Cams of programming switches PS2 and PS3 are spring loaded (on the shaft) to an open neutral position. If the Inboard Whip winch drum rotates, indicating movement of the trolley relative to the delivering ship, through flexible shaft FS1, gearing, and engaged clutch CL1 the cams of programming switches PS2 and PS3 are also rotated.

If the trolley moves a very small distance towards the delivering ship programming switch PS2 is closed completing the electric circuit branch K1. Solenoid SOL 8 is energized. The spool of valve DSV1 is shifted from its spring loaded center position A to position B. Hydraulic flow through valve DSV1 to the A end and from the B end of stop correction control actuator CA3 advances the rod of CA3 to stroke simultaneously variable pumps PV3 and PV4 to "Deliver."

As the trolley moves away from the delivering ship, the cam of programming switch PS2 is rotated in the opposite direction. As the trolley reaches its desired position (the position desired when the control operator stopped the trolley), programming switch PS2 is opened. Solenoid SOL8 is energized and the spool of valve DSV1 returns to spring loaded center position A blocking all valve ports at that valve.

If the trolley moves a very small distance away from the delivering ship, a similar sequence of events occurs ultimately adjusting the stroke of transfer variable pumps PV3 and PV4 to return the trolley toward the delivering ship a slight amount. PS3 is closed, solenoid SOL9 is energized shifting the spool of valve DSV1 to position C, with resultant hydraulic flow to the B end and from the A end of control actuator CA3. Thus the strokes of transfer variable pumps PV3 and PV4 are simultaneously adjusted toward the return direction.

5.352 <u>If phased to the receiving ship</u>, programming switch PS1 is closed and current flows through electric circuit branch D2. Relay R1 is energized: relay operated contacts RC11 are open while relay

operated contacts RC12 are closed. Hence branch P2 is closed while branch P1 is open. Clutch CL2 is engaged therefore the cams of programming switches PS2 and PS3 are driven through gears by the cross shaft of the differential. This shaft, as further explained below, rotates as the trolley position changes, as related to the receiving ship.

The stop correction system operates in a manner similar to that described above but with the stopped position of the trolley related to the receiving ship instead of the delivering ship. If movement of the trolley toward the receiving ship is detected programming switch PS3 is closed; if movement of the trolley away from the receiving ship is detected, programming switch PS2 is closed.

5.4 Automatic Transfer Control

5.41 Place operator switch OS2 in "Automatic" position allowing current flow in electric circuit branch D4 from F to H. Solenoids SOL6 and SOL7, electric circuit branch H2, are energized. Valves SS3V5 and SS3V6 have their spools shifted from spring loaded positions A to positions B permitting flow to and from valve DSV2 through valves SS3V5 and SS3V6 while isolating servo valve SV3.

5.42 To "Stop," place operator switch OS3 in "Stop" position (this switch is spring loaded to "stop" position and held in "Deliver" or "Return" positions by energized solenoids SOLA or SOLB respectively). With operator switch OS3 in the "Stop" position current flows through electric circuit branch H31 energizing relay R2. Energization of relay R2 closes relay operated contacts RC21 through 25 and opens relay operated contacts RC26 through 28.

Control response will depend upon the position of the transfer control linkage at the time at which operator switch OS3 is shifted to the "Stop" position.

5.421 If the transfer control linkage is at or near its center "Stop" position, normally closed snap acting switches SA3 and SA5 will be held open by cam surfaces on the transfer control linkage. Electrical branches I and II are broken: there is no current flow through relays R5 and R6 or solenoids SOL10 or SOL11. The normally closed relay contacts RC51, 52, 61 and 62 are closed. Thus electric circuit branch D4 is closed to point K and electric circuit branch T2 is also closed. The preceding description of the stop correction system applies.

5.422 If the transfer control linkage is displaced towards the "Deliver" direction at the time at which operator switch OS3 is shifted to the "Stop" position, snap acting switch SA3 is disengaged from its cam and is in the normally closed position. Current flows through electric circuit branch I energizing relay R5 and solenoid SOL11. Normally closed relay operated contacts RC51 and 52 are opened by relay R5 thereby prohibiting operation of the stop correction system. Energization of solenoid SOL11 shifts the spool of valve DSV2 to position C. Servo hydraulic flow through valves DSV2 and SS3V6 to the B end of transfer control actuator CA2 moves the rod of control actuator CA2 with the transfer control linkage in the "Return" direction. The A end of transfer control actuator CA2 is drained through valves SS3V5 and DSV2 and restriction RE2, which impedes the return flow thus limiting trolley deceleration to 12 fps^2. As the transfer control linkage reaches its center "stop" position, normally closed snap acting switch SA3 is engaged by its cam and opened discontinuing current flow through branch I. Relay R5 and solenoid SOL11 are de-energized. The spool of valve DSV2 shifts from position C to its spring loaded center position A thereby blocking hydraulic flow to and from the transfer control actuator CA2. Relay operated contacts RC51 and 52 shift to their normally closed positions allowing current flow in branches H1 and T2. The transfer control linkage is now at or near its center "stop" position. The stop correction system functions as described above.

5.423 If the transfer control linkage is displaced towards the "Return" direction at the time at which operator switch OS3 is shifted to the "stop" position, snap acting switch SA5 is disengaged from its cam and is in the normally closed position. Current through branch II energizes relay R6 and solenoid SOL10. Energization of relay R6 opens relay operated contacts RC61 and 62 prohibiting operation of the stop correction system. Energization of solenoid SOL10 shifts the spool of valve DSV2 to position B thereby moving the transfer control linkage towards the "Deliver" direction. As the transfer control linkage reaches its center "Stop" position, normally closed snap acting switch SA5 is engaged by its cam and opened discontinuing current flow through branch II. Relay 6 and solenoid SOL11 are de-energized. The spool of valve DSV2 shifts from

position B to its spring loaded center position A. Relay operated contacts RC61 and 62 shift to their normally closed positions. The transfer control linkage is now at or near its center "stop" position. The stop correction system functions as described above.

5.43 To "Deliver", place operator switch OS3 in the "deliver" position. Electric circuit branch H32 to 0 is completed. Solenoid SOLA and relay R3 are energized. Energization of solenoid SOL A maintains operator switch OS3 in the "Deliver" position. Energization of relay R3 closes relay operated contacts RC31 and opens normally closed relay operated contacts RC32. Resultant current flow through branch D6-III energizes relay R6 and solenoid SOL10. (Operation of relay R6 concerns the stop correction system and was explained above.) Energization of solenoid SOL10 shifts the spool of valve DSV2 from spring loaded center position A to position B. Servo hydraulic flow, through valves DSV2 and SS3V5 is to the A end of the transfer control actuator CA2 which moves the transfer control linkage towards the "Deliver" direction. The B end of the transfer control actuator CA2 is drained through valves SS3V6 and DSV2 and restriction RE2 which impedes the return flow thus limiting trolley acceleration to 12 $fps.^2$ As the transfer control linkage reaches its maximum speed "Deliver" position, normally closed snap acting switch SA2 is engaged by its cam and opened discontinuing current flow through branch III. Relay R6 and solenoid SOL10 are de-energized. The spool of valve DSV2 shifts from position B to its spring loaded center position A thereby blocking hydraulic flow to and from the transfer control actuator CA2. The trolley is now proceeding in the "Deliver" direction at maximum speed.

As the trolley proceeds from the Delivering Ship toward the Receiving Ship, flexible shafts FS1 and FS2 common with the Inboard Whip winch drum and the Transfer Whip winch drum shafts directly drive, through suitable gears, the sun gears of a differential at the Transfer Sensor. The resulting revolutions of the differential cross shaft are indicative of trolley position as related to the Receiving Ship. The differential cross shaft is coupled directly to a pinion P2 which meshes with a spiroid gear SG2 which in turn rotates the cam of programming switch PS4. The rotational speed of the differential cross shaft is reduced 50:1 through the pinion-spiroid gear P2-SG2 combination.

When the trolley reaches point II, programming switch PS4 is cam operated from its normally closed position to open. The electrical circuit branch H32 is broken de-energizing solenoid SOL A and relay R3. Relay operated contacts

RC31 return to their normally opened position while relay operated contacts RC32 return to their normally closed position. De-energization of solenoid SOLA releases operator switch OS3 which shifts to its center spring loaded "stop" position. The trolley is brought to a stop in the manner described above with the transfer control linkage displaced towards the "Deliver" direction at the time at which operator switch OS3 is shifted to the "stop" position, paragraph 5.422.

5.44 To "Return", place operator switch OS3 in "Return" position. Electric circuit branch H33 to O is completed. Solenoid SOLB and relay R4 are energized. Energization of solenoid SOLB maintains operator switch OS3 in the "Return" position. Energization of relay R4 closes relay operated contacts RC41 and opens normally closed relay operated contacts RC42. Resultant current flow through branch D2-IV energizes relay R5 and solenoid SOL11. (Relay R5 concerns the stop correction system.) Energization of solenoid SOL11 shifts the spool of valve DSV2 from spring loaded center position A to position B. Servo hydraulic flow through valves DSV2 and SS3V6 into the B end of the transfer control actuator CA2 which moves the transfer control linkage towards the "Return" direction. The A end of the transfer control actuator CA2 is drained through valves SS3V5 and DSV2 and restriction RE2 which impedes the return flow thus limiting trolley acceleration to 12 fps^2. As the transfer control linkage reaches its maximum speed "Return" position, normally closed snap acting switch SA4 is engaged by its cam and opened discontinuing current flow through branch IV. Relay R5 and solenoid SOL11 are de-energized. The spool of valve DSV2 shifts from position C to its spring loaded center position A thereby blocking hydraulic flow to and from the transfer control actuator CA2. The trolley is now proceeding in the "Return" direction at maximum speed.

As the trolley proceeds from the Receiving Ship toward the Delivering Ship, flexible shaft FS1 common with the Inboard Whip winch drum drives a pinion P1 which meshes with a spiroid gear SG1. The spiroid gear SG1 in turn rotates the cam of programming switch PS5. The rotational speed of the flexible shaft FS1 is reduced 50:1 through the pinion-spirod gear P1-SG1 combination.

When the trolley reaches point III, programming switch PS5 is cam operated from its normally closed position to open. The electrical circuit branch H33 is broken de-energizing solenoid SOLB and relay R4. Relay operated contacts RC41 return to their normally opened position while relay operated contacts RC42 return to their normally closed position. De-energization of solenoid SOLB releases operator switch OS3

which shifts to its center spring loaded "stop" position. The trolley is brought to a stop in the manner described above with the transfer control linkage displaced towards the "Return" direction at the time at which operator switch OS3 is shifted to the "stop" position, paragraph 5.423.

6. SLIDING BLOCK (Figures 2 and 6)

6.1 To raise the sliding block the operator depresses normally open spring loaded push button switch OS4 completing electric circuit branch D8. Solenoids SOL12 and SOL14 are energized. The spool of valve DSV3 is shifted from its spring loaded center position A to position B, and the spool of valve SS3V7 is shifted from its spring loaded position A to position B.

Servo hydraulic flow through valve DSV3 to and from fixed hydraulic motor MF3 drives a roller chain sprocket to raise the sliding block.

Restriction RE3 impedes the return hydraulic flow thereby limiting the sliding block acceleration and speed as desired.

Replenishing hydraulic flow through valve SS3V7 releases spring set brake BR3 which otherwise would tend to hold the sliding block at a fixed location.

The operator maintains manual pressure upon push button switch OS4 so long as he desires the sliding block to continue to raise.

If the upper limit of travel of the sliding block is reached normally closed limit snap acting switch SA6 is cam operated to open and electrical circuit Branch D8 is broken thereby de-energizing solenoids SOL12 and SOL14.

The spools of valves DSV3 and SS3V7 shift from positions A to their spring loaded positions B. Hydraulic flow to and from fixed hydraulic motor MF3 is blocked by valve DSV3 and the hydraulically released brake BR3 is isolated and drained by valve SS3V7. Brake BR3 springs to its locked position. The upward travel of the sliding block is stopped and locked. To stop the sliding block at an intermediate height the control operator releases switch OS4; sequence of operations is the same as if the upper limit was reached except that electric circuit branch D3 is broken by the open switch OS4 instead of the limit switch SA6.

6.2. To lower the sliding block the operator depresses normally open spring loaded push button switch OS5. Sequence of operations is similar to that described above for raising the sliding block. With switch OS5 depressed solenoids SOL13 and SOL14 are energized. The spool of valve DSV3 shifts from spring loaded position A to

position B establishing hydraulic flow to and from fixed hydraulic motor MF3 thereby lowering the sliding block. The spool of valve SS3V7 shifts from spring loaded position A to position B, thereby releasing brake BR3 freeing the sliding block for operation.

If the lower limit of travel of the sliding block is reached normally closed limit snap acting switch SA5 is opened thereby de-energizing solenoids SOL 13 and SOL 14. The spools of valves DSV3 and SS3V7 shift from positions C and B, respectively, to positions A. Hydraulic flow to and from the fixed hydraulic motor MF3 is blocked and brake BR3 is set. The downward travel of the trolley is thus stopped.

By releasing push button switch OS5 the operator can stop the sliding block at any intermediate position.

7. STATUS, TENSIONED MODIFIED HOUSEFALL MODEL MARK II.

7.1 Plans developed for Tensioned Modified Housefall Model Mark II are:

BUSHIPS NO.	Title
521 2022315A	TMHF Model MK.II Hydraulic Motor and Winch Drum Shafts and Mounts.
521 2022328	TMHF Model MK.II Preliminary Arrangement and Structural Details, Sliding Block.
521 2022330	TMHF Model MK.II Hydraulic Pump and Brake Mounts.
521 2022384	TMHF Model MK.II Hydraulic Manifold

7.2 Items on hand at San Francisco Naval Shipyard

Item	Quantity	Remarks and/or Description
MF1 & 2	2	WSI Series 900 Fixed Displacement Hydraulic Motor. See page 66 of reference (a) for description. On loan from Washington Scientific Industries, Inc., Minneapolis, Minnesota.
PV1 thru 4	4	Vickers PVB-10 Variable Displacement Hydraulic Pump. See page 66 of reference (a) for description.
PV5	1	Vickers PVB-5 Pressure Compensated Variable Displacement Pump. See page 66 of reference (a) for description.

Item	Quantity	Remarks and/or Description
PV6	1	Vickers VV3-50-10 Pressure Compensated Variable Displacement Pump. See page 66 of reference (a) for description.
EM1	1	Marathon Electric 15 HP AC Motor, 1760 RPM, TEFC, Double Shaft. See page 67 of reference (a) for description.
EM2 & 3	2	Marathon Electric 5HP AC Motor, 1740 RPM, TEFC, Double Shaft. See page 67 of reference (a) for description.
Controller for EM1	1	440 Volt 3 Phase 60 cps AC
Controller for EM 2 & 3	2	440 Volt 3 Phase 60 cps AC
Push Button Start-Stop switches	3	NEMA Watertight enclosures; for EM1 thru 3.
Winch Drum	1	Transfer Whip. LeBus grooved for 1/8" 7 x 19 strand core CRES wire.
Winch Drum	1	Inboard Whip. LeBus grooved for 1/8" 7 x 19 strand core CRES wire.
Pressure Roller Assemblies	2	LeBus
Shaft Couplings	8	Thomas Flexible Disc Type. MF1 and 2 to winch drum shafts EM1 to PV1 & 2 EM2 to PV3 & 4 EM3 to PV5 & 6
Flange Mounted Bearings	4	Schafer Self Aligned to support winch drum shafts.
Sheaves	18 approx.	Tapered Roller Bearing 5" & 6" nominal OD. Grooved for 1/8" wire. To be used for sliding block assembly, fair leads, trolley and housefall block.

Item	Quantity	Remarks and/or Description
Wire	1000 ft. approx.	1/8" 7 x 19 strand core CRES.

7.3 The main hydraulic manifold was designed to mount the following valves (BUSHIPS Plan 521 2022364).

Item	Quantity	Remarks and/or Description
PHV 1 & 2	2	Rivett Lathe & Grinder, Inc. model 6805-1. Double hydraulic pilot operated. No springs, 2 position, 4-way valve (to be used as 3-way valve). 1" size. Mounting surfaces to be machined on manifold.
RV1 thru 4	4	Fawick Corporation Hydraulic-Electronic Division adjustable cartridge relief valve 131189-020. Mounting cavities to be machined in manifold.
CV1 thru 4	4	Combination pump Valve Co. check valve cartridges catalog No. 665-1. Mounting cavities to be machined in manifold.
M4V1	1	Denison valve 1/4" 4-way manually operated detent positioned, sub-plate type. 3 detent No. DD 023 513 D type 3 spool (to be used in 2 spool positions only, including center position). Mounting surface to be machined on manifold.

7.4 The following items are required. No commitment exists in design or availability for specific manufacturers' components listed. This is not a complete list.

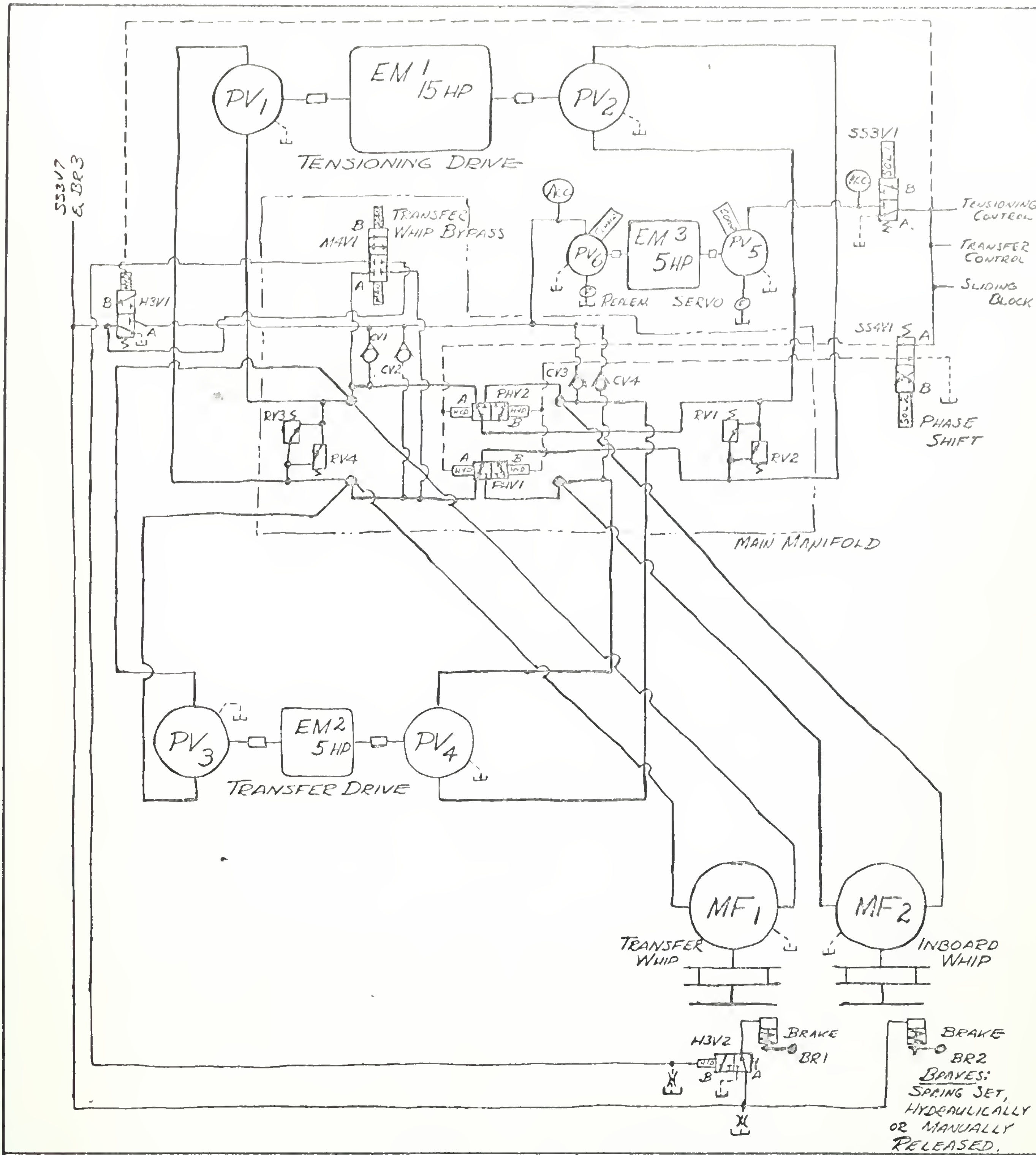
Item	Quantity	Remarks and/or Description
OS1 & 2	2	Manually operated 2-position detented switch.
OS3	1	Manually operated 3-position switch. Spring loaded to center "Stop" position. Held in "Deliver" position by energized solenoid SOL A. Held in "Return" position by energized solenoid SOL B.
OS4 & 5	2	Manually operated push button switches. Spring loaded open.

Item	Quantity	Remarks and/or Description
SOL A & B	2	Maintain OS3 in "Deliver" or "Return" position when energized.
SOL 1 thru 14	14	Valve actuators. Obtained with valves. These solenoids should be heavy duty oil immersed type. 115 volts 60 cps A.C.
PS 1 & 5, PS 2 & 3	2 assemblies 2 programming switches each assembly	Precision Mechanisms Corp. Adjustable cam switch model CS 402-2 [illegible]
PS4	1	Precision Mechanisms Corp. Adjustable cam switch model CS402-1 with double ended shaft extension.
SA1 thru 7	7	Salt water and oil environment proof snap acting switches.
R1, RC11 & 12 R2, RC21 thru 28 R3, RC31 & 32 R4, RC41 & 42 R5, RC51 & 52 R6, RC61 & 62	6	Salt water and oil environment proof relays.
CL1 & 2	1 assembly	Guidance Controls Corp. Duplex Clutch Model HCC 8 24-28 Volt DC.
Transformer-Rectifier	1	115 Volts 60 cps AC to 28 Volts DC for clutch operation.
TX1 & 2	2	Synchro Transmitters to provide remote indication of trolley distance to either replenishing ship.
Synchro Receivers	2	Not shown. To provide remote indication of Trolley distance to either replenishing ship.

Item	Quantity	Remarks and/or Description
FS1 & 2	2	Mechanical signal transmission from winch drums to Transfer Sensor. Should be large size to minimize lost motion.
P1-SG1, P2-SG2	2 sets	Precision Instrument Components catalog code EJ & EK. Spiroid gears.
Differential	1	Precision Instrument Components catalog code V.
SV1 thru 3	3	Mechanical servo valves. Will most likely require specific designs. Size depends upon control dynamics.
CA1 thru 3	3	Double end double acting hydraulic actuators, 2000 psi. rating. Size dependent upon avilability and control dimensions and dynamics.
DA1	1	Single end double acting hydraulic actuator, 2000 psi. Size dependent upon avilability and control dimensions and dynamics.
DSV1 thru 3	3	Valve 1/4", 4-way, 3-position, double solenoid operated, spring centered, heavy duty oil immersed solenoids.
SS4V1	1	Valve, 1/4", 4-way, 2-position, single solenoid operated, heavy duty oil immersed solenoid.
SS3V1 thru 6	6	Valve, 1/4", 3-way, 2-position, single solenoid operated, heavy duty oil immersed solenoid.
H3V1 & 2	2	Valve, 1/4", 3-way, 2-position, single hydraulic pilot operated.
H2V1 & 2	2	Valve, 1/4", 2-way, 2-position, single hydraulic pilot operated.
RE1 thru 3	3	For initial orifice determination: variable calibrated orifices.
MF3	1	Low speed high torque fixed displacement hydraulic motor. Washington Scientific Industries, Inc. Motor Series 350 probably well suited.
F	2	Filters. Initial and continuous filtration to 25 microns or less is essential.

Item	Quantity	Remarks and/or Description
Acc	2	Replenishing system: 200 psi. Servo System 1500-2000 psi.
BR 1 thru 3	3	BuShips plan No. 521 2022330 based upon Goodyear Industrial brake No. PD 1436 SK 1. Wagner Electric Corp. Type SOH industrial brake also adaptable.

APPENDIX



REV.	DESCRIPTION	DATE	APP'D
	REVISIONS		

REPLENISHMENT-AT-SEA
CONTRACT GUIDANCE DRAWING

APPROVED	DATE
	DESIGN SUPT.
OFFICIAL	SIGNATURE
HD. ENGR.	
BR. SUP.	
SEC. SUP.	
REVIEWED	
DRAWN	SHILLINGER

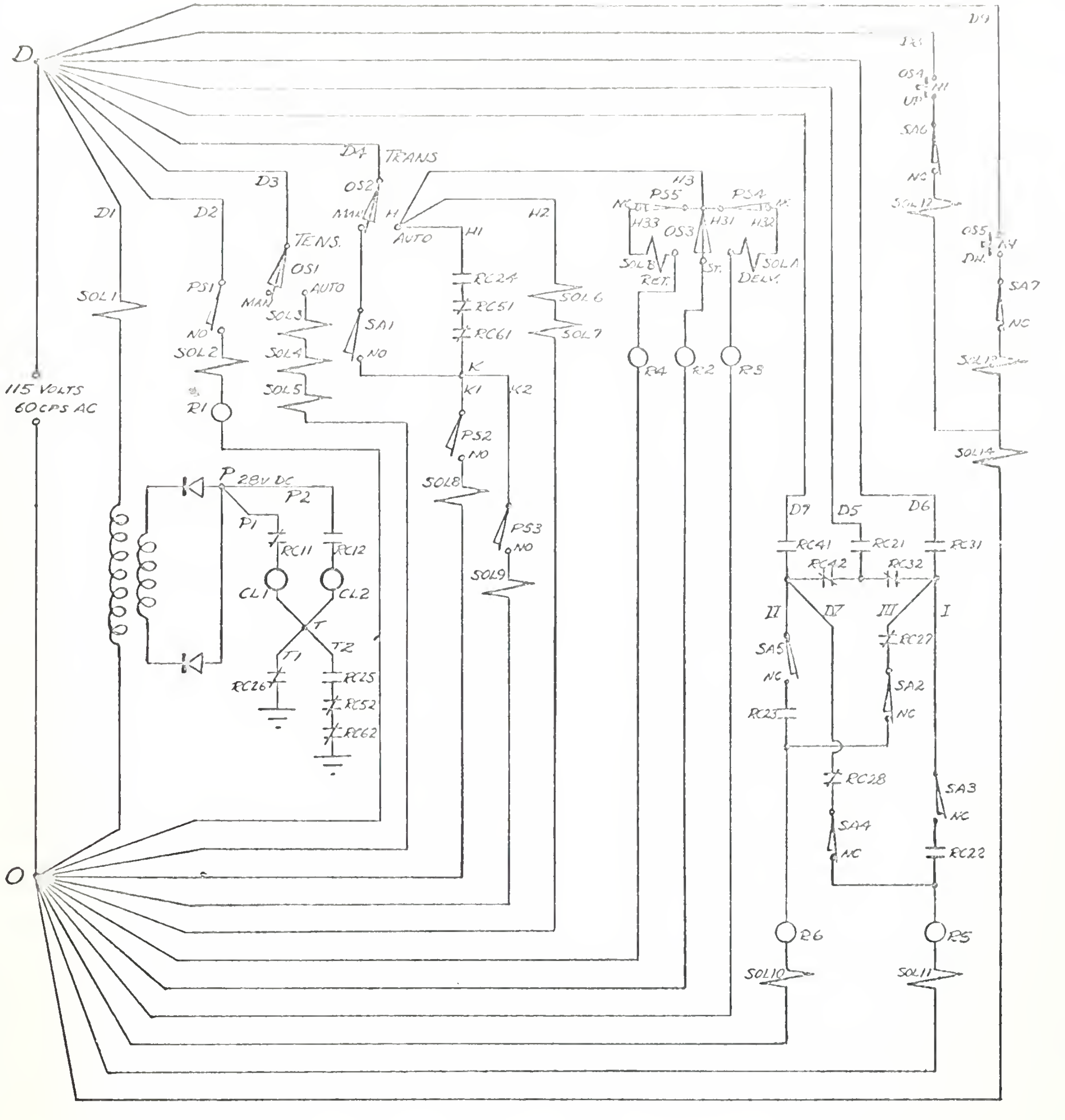
SAN FRANCISCO NAVAL SHIPYARD
SAN FRANCISCO 24, CALIFORNIA

FIGURE 1

MAIN DRIVE HYDRAULIC CIRCUIT

BUREAU OF SHIPS NO. | REV.

SCALE | SHEET | OF



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				OFFICIAL	SIGNATURE
				HD. ENGR.	
				BR. SUP.	
REV.				SEC. SUP.	
	DESCRIPTION	DATE	APP'D	REVIEWED	
	REVISIONS			DRAWN	SHILLINGER

REPLENISHMENT-AT-SEA
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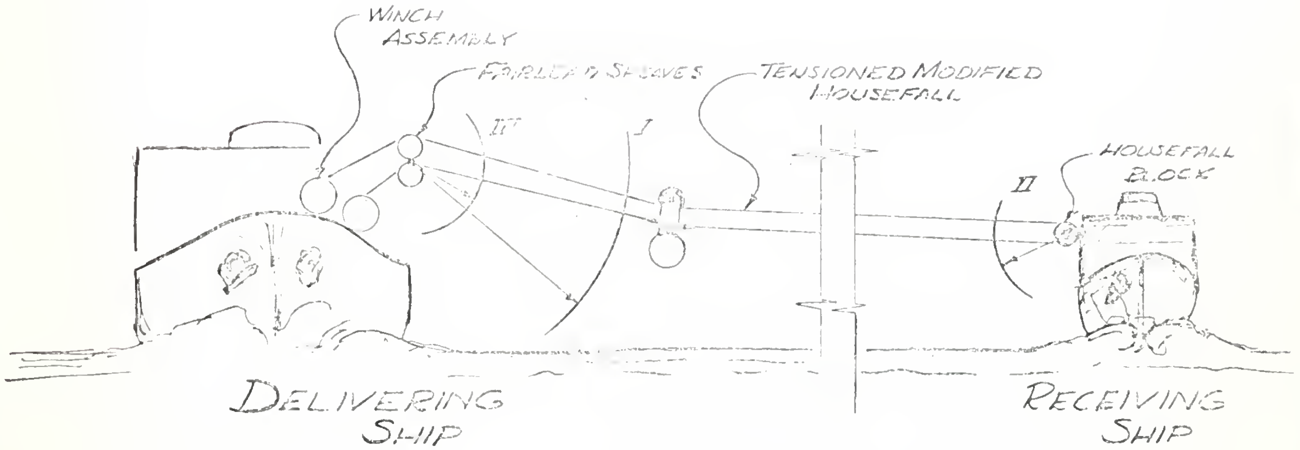
SAN FRANCISCO 24, CALIFORNIA

FIGURE 2

ELECTRIC CONTROL CIRCUIT

BUREAU OF SHIPS NO.			REV.
SCALE		SHEET	OF



POINTS

I	PHASE SHIFT
II	DECELERATION POINT, DELIVER DIRECTION
III	DECERATION POINT, RETURN DIRECTION

REV.	DESCRIPTION	DATE	APP'D	
REVISIONS				

APPROVED	DATE
	DESIGN SUPT.
OFFICIAL	SIGNATURE
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BR. SUP.	
SEC. SUP.	
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REPLENISHMENT-AT-SEA
CONTRACT GUIDANCE DRAWING

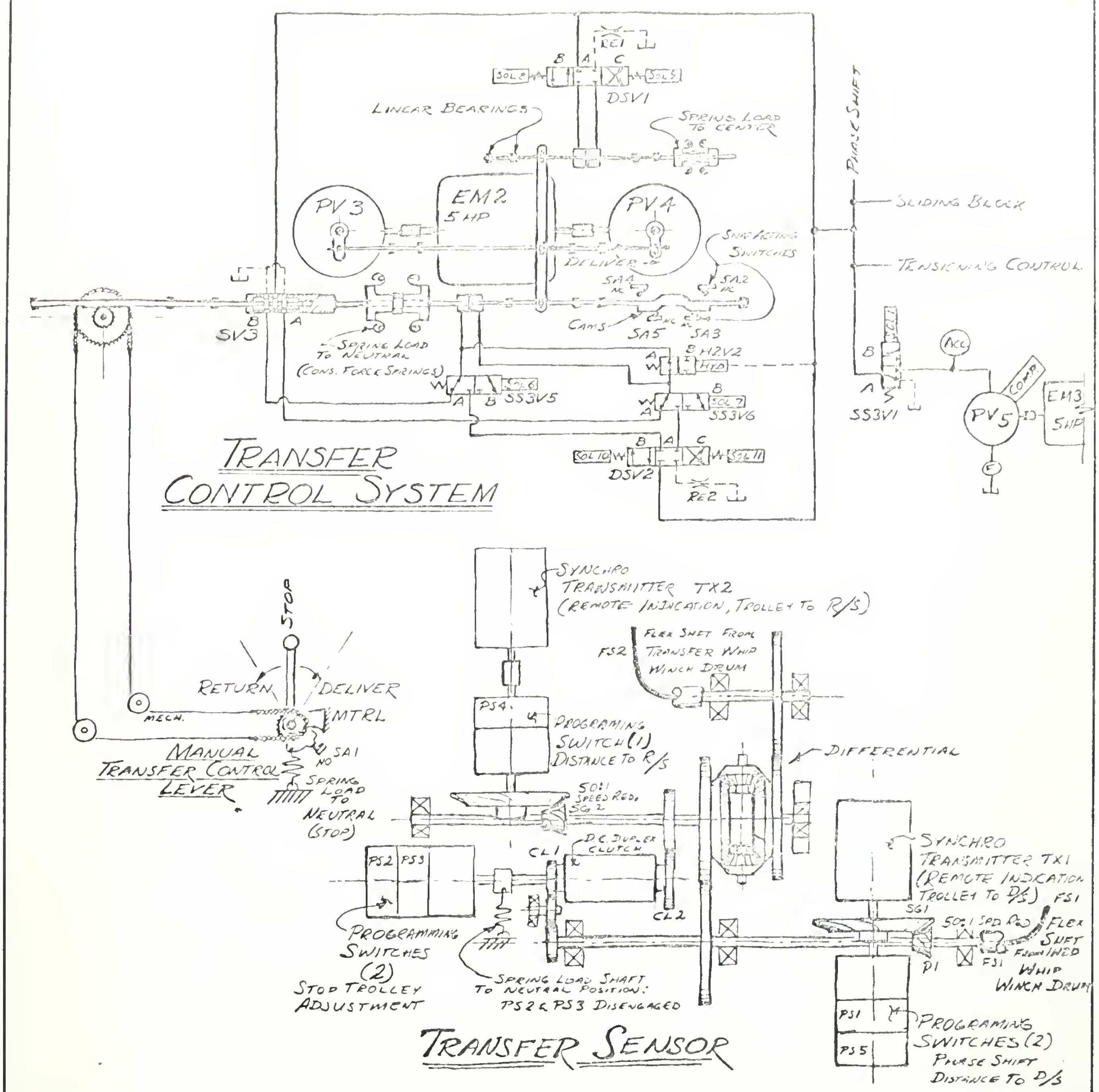
SAN FRANCISCO NAVAL SHIPYARD
SAN FRANCISCO 24, CALIFORNIA

FIGURE 3

PHASE SHIFT AND
DECELERATION POINTS

BUREAU OF SHIPS NO. | REV.

SCALE | SHEET | OF



REV.	DESCRIPTION	DATE	APP'D
	REVISIONS		

APPROVED DATE

DESIGN SUPT.

OFFICIAL SIGNATURE

HD. ENGR.

BR. SUP.

SEC. SUP.

REVIEWED

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SAN FRANCISCO 24, CALIFORNIA

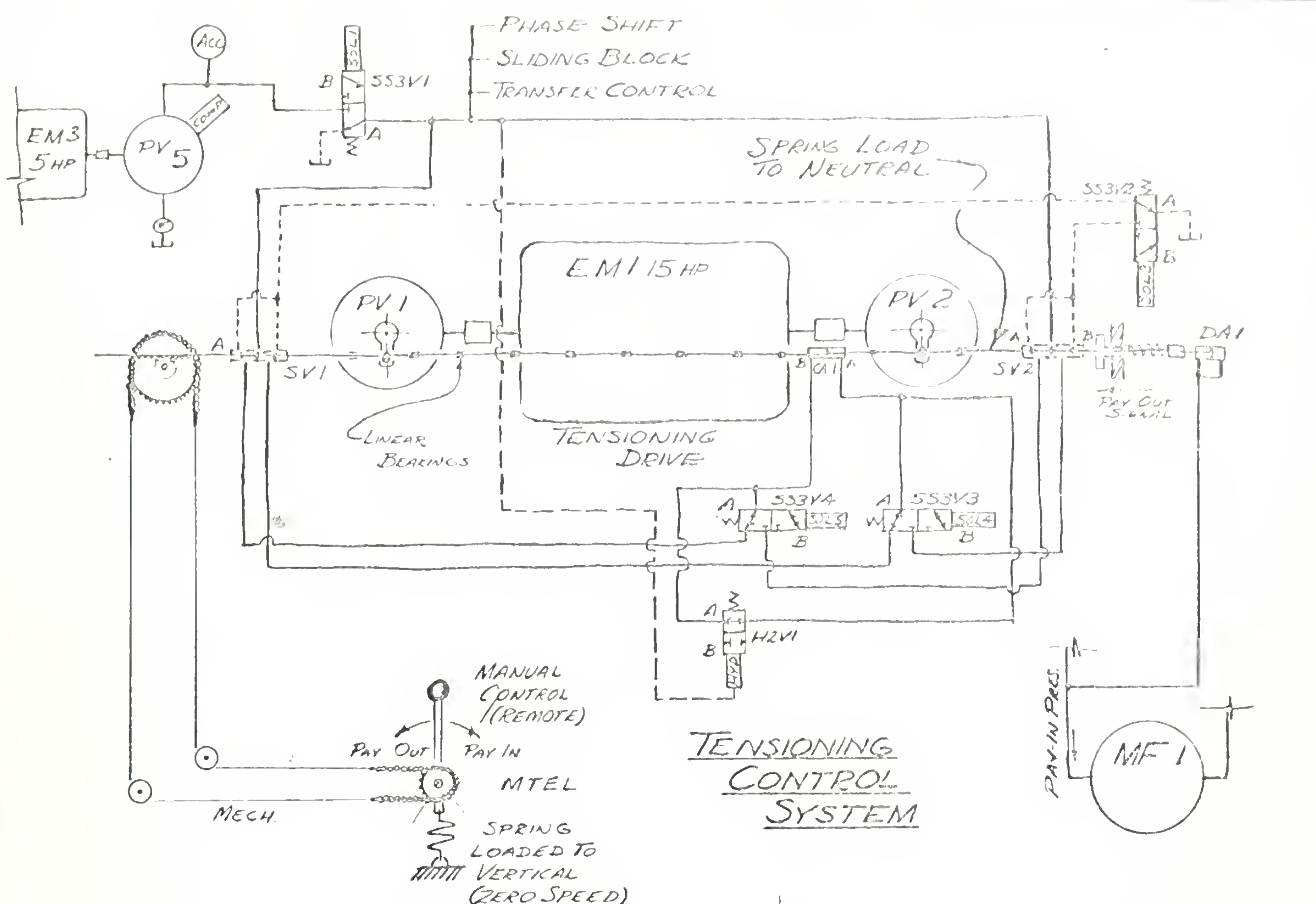
FIGURE 4

TRANSFER SENSOR AND TRANSFER CONTROL SYSTEM

BUREAU OF SHIPS NO. REV.

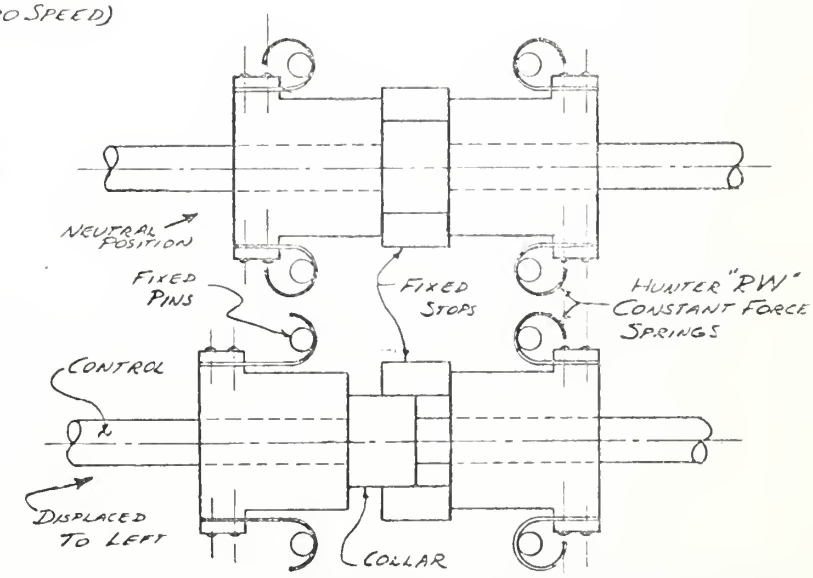
SCALE SHEET OF

REPLENISHMENT-AT-SEA CONTRACT GUIDANCE DRAWING



TENSIONING CONTROL SYSTEM

METHOD OF SPRING LOADING TO NEUTRAL



REV.	DESCRIPTION	DATE	APP'D
	REVISIONS		

REPLENISHMENT-AT-SEA
CONTRACT GUIDANCE DRAWING

APPROVED	DATE
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HD. ENGR.	
BR. SUP.	
SEC. SUP.	
REVIEWED	
DRAWN	SHILLINGER

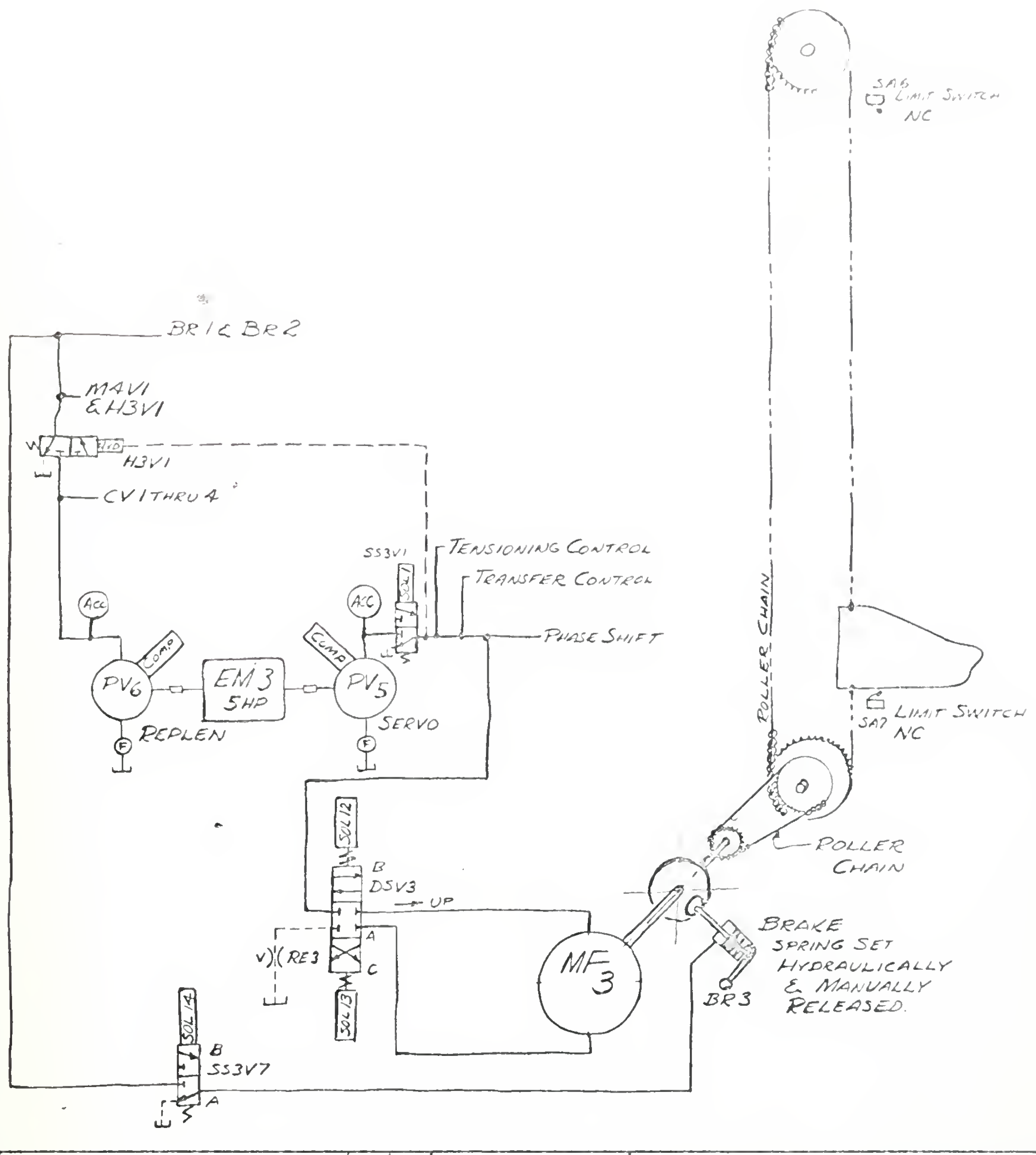
SAN FRANCISCO NAVAL SHIPYARD
SAN FRANCISCO 24, CALIFORNIA

FIGURE 5

SPRING LOADING AND TENSIONING CONTROL SYSTEM

BUREAU OF SHIPS NO. REV.

SCALE SHEET OF



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				HD. ENGR.		
				BR. SUP.		
REV.				SEC. SUP.		
	DESCRIPTION	DATE	APP'D	REVIEWED		
	REVISIONS			DRAWN	SHILLINGER	

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FIGURE 6

SLIDING BLOCK CONTROL SYSTEM

BUREAU OF SHIPS NO. | REV.

SCALE | SHEET | OF

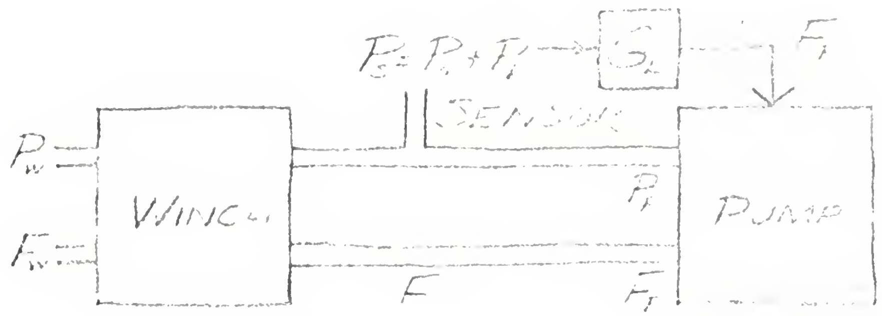
REPLENISHMENT-AT-SEA
CONTRACT GUIDANCE DRAWING

APPENDIX E

LUMPED IMPEDANCE ANALYSIS

LUMPED IMPEDANCE ANALYSIS

(CHAP 4 ME 134 HANDOUT NOTES)



$$\begin{bmatrix} P_W \\ F_W \end{bmatrix} = \begin{bmatrix} 1 & Ls \\ 0 & 1 \end{bmatrix} \begin{bmatrix} A_v\left(\frac{M}{A_v} \cdot M_f A_f s\right) \\ 0 & \frac{1}{A_v} \end{bmatrix} \begin{bmatrix} 1 & \frac{R_f}{2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 \\ \frac{A_p l}{B} s & 1 \end{bmatrix} \begin{bmatrix} 1 & \frac{R_f}{2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 \\ \frac{A \Delta Y}{E(\frac{D_o}{D_i}-1)} & 1 \end{bmatrix} \begin{bmatrix} P_S \\ F \end{bmatrix}$$

INERTIA — MECH-HYDR PISTON & LIQ INERTIA — FRICTION & COMPLEX. — PIPE WALL ELAST.

$$\begin{bmatrix} P_W \\ F_W \end{bmatrix} = \begin{bmatrix} a & b \\ c & d \end{bmatrix} \begin{bmatrix} P_S \\ F \end{bmatrix} \quad \text{---}(1)$$

$$\begin{bmatrix} P_W \\ F_W \end{bmatrix} = \begin{bmatrix} a & b \\ c & d \end{bmatrix} \begin{bmatrix} 1 & \frac{R_f}{2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 \\ \frac{A_p l}{F} s & 1 \end{bmatrix} \begin{bmatrix} 1 & \frac{R_f}{2} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} 1 & 0 \\ \frac{A \Delta Y}{E(\frac{D_o}{D_i}-1)} & 1 \end{bmatrix} \begin{bmatrix} P_T \\ F_T \end{bmatrix}$$

$$\begin{bmatrix} P_W \\ F_W \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} P_T \\ F_T \end{bmatrix} \quad \text{---}(2)$$

$$\left.\begin{aligned} P_w &= aP_s + bF \\ F_w &= cP_s + dF \end{aligned}\right\} \text{EXPAND (1)}$$

$$\left.\begin{aligned} dP_w &= adP_s + bdF \\ bF_w &= bcP_s + bdF \\ dP_w - bF_w &= (ad-bc)P_s \end{aligned}\right\} \text{ELIMINATE } F$$

$$P_s = \frac{dP_w - bF_w}{ad-bc}$$

$$\left.\begin{aligned} P_w &= AP_p + BF_p \\ F_w &= CP_p + DF_p \end{aligned}\right\} \text{EXPAND (2)}$$

$$\left.\begin{aligned} AF_w &= ACP_p + ADF_p \\ CP_w &= ACP_p + BCF_p \\ AF_w - CP_w &= (AD-BC)F_p \end{aligned}\right\} \text{ELIMINATE } P_p$$

$$\left.\begin{aligned} F_p &= G_2 P_s = G_2\left(\frac{dP_w - bF_w}{ad-bc}\right) \\ AF_w - CP_w &= (AD-BC)G_2\left(\frac{dP_w - bF_w}{ad-bc}\right) \end{aligned}\right\} \text{SUBSTITUTE}$$

$$\text{LET } H = \frac{G_2(AD-BC)}{ad-bc}$$

$$AF_w - CP_w = HdP_w - HbF_w$$

$$(A+Hb)F_w = (C+Hd)P_w$$

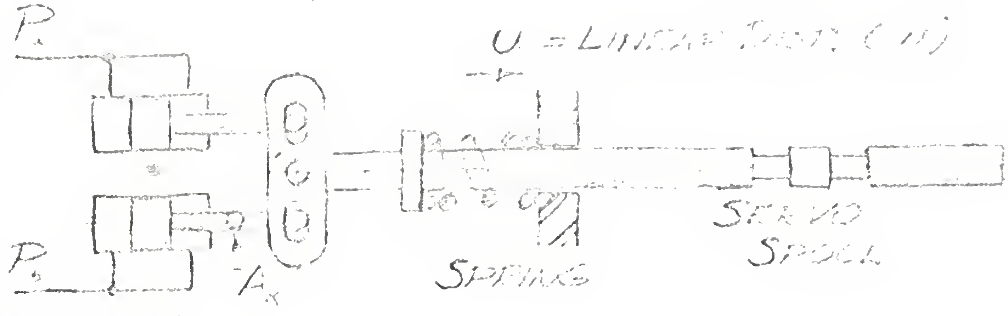
$$G_{①} = \frac{F_w}{P_w}$$

$$G_{①} = \frac{C+Hd}{A+Hb}$$

$$G_L = \frac{K_2 K_3 G_5 K_1}{1 + K_4 G_5}$$

ASSUME INERTIA, FRICTION SATURATION = 0
FOR CONTROL LINKAGES

$$\xrightarrow[\text{PSI}]{P_s} \boxed{K_3} \xrightarrow[\text{IN.}]{U}$$



$P_s = P_a + P_b$

$$(P_a + P_b) A_R = k_s U$$

$$P_s A_R = k_s U$$

$$K_3 = \frac{U}{P_s}$$

$$K_3 = \frac{A_R}{k_s}$$

WHERE:

A_R = PISTON ROD AREA, (.0491 SQ IN)

k_s = SPRING RATE, LBS./IN.
(475 LBS/IN. SUITABLE)

$$K_3 = \frac{.0491}{475}$$

$$K_3 = 1.03 \times 10^{-4} \text{ IN/PSI}$$

$$\xrightarrow[\text{IN.}]{U-V} \boxed{K_4} \xrightarrow[\text{CIS}]{\Gamma}$$

FROM <u>FLUID [illegible]</u> [illegible] PRETHOF & CHEANEY, PAGE 161:

$$q_{s} = C_d a \sqrt{\frac{2p}{\rho}}$$

WHERE:

q = RATE OF FLOW CU. IN. [illegible], CIS

a = AREA OF OPENING, SQ IN

p = PRESSURE DROP ACROSS OPENING

ρ = FLUID DENSITY, LB/CU IN

C_d = DISCHARGE COEFFICIENT

ASSUME REYNOLDS NUMBER > 260 (CRITICAL VALUE)

$C_d = 0.60$ APPROX. FOR SHARP-EDGE CONTROL VALVE

$a = \frac{1}{8} \cdot \frac{\pi}{4} = \frac{\pi}{32}$ SQ IN. (STROKE $= \frac{1}{8}''$, LINE DIA $= \frac{1}{4}''$)

$\rho = 0.0304$ LB/CU IN. (INDUST PET. HYDR OIL)

$p = 1000$ PSI (ASSUMED)

$$q_{(s)} = (0.60)\frac{\pi}{32}\sqrt{\frac{2(1000)}{.0304}}$$

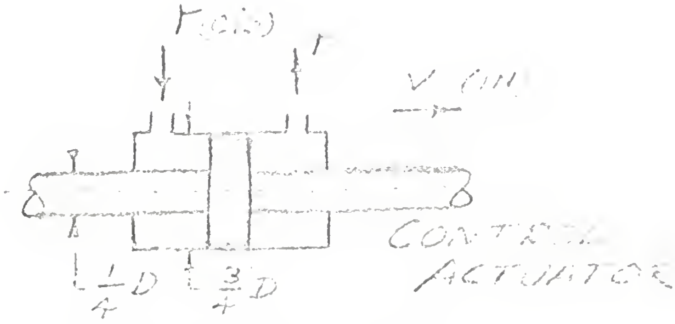
$q_{(s)} = 15.15$ CIS FOR $\frac{1}{8}''$ STROKE

$\Gamma = q_{(s)}$ WHERE $U-V = \frac{1}{8}''$

$$K_4 = \frac{\Gamma}{U-V} = \frac{15.15}{.125}$$

$$K_4 = 121.2 \text{ CIS/IN.}$$

ASSUME HYDRAULIC FLOW VARIES DIRECTLY WITH LINEAR DISPLACEMENT OF SERVO SPOOL.



$$A_n = \frac{\pi}{4}\left[\left(\frac{3}{4}\right)^2 - \left(\frac{1}{4}\right)^2\right]$$

$$A_n = \frac{\pi}{8} \text{ SQ. IN.}$$

$$v = f(r, A_n)$$

WHERE: A_n = ANNULUS AREA, SQ IN.

$$\frac{v}{dt} = vS = \frac{r}{A_n}$$

$$G_{(S)_5} = \frac{v}{r}$$

$$G_5 = \frac{1}{A_n S}$$

$$G_5 = \frac{8}{\pi S}$$

$$G_5 = \frac{2.55}{S} \quad \text{IN}/\text{CIS}$$

ASSUME:

INERTIA, FRICTION, STICTION, LEAKAGE = 0

$$\xrightarrow[\text{IN.}]{V} \boxed{K_6} \xrightarrow[\text{CIS.}]{F_P}$$

$$F_{P(MAX)} = 1700\,\text{RPM} \times \frac{1}{60}\frac{\text{SEC}}{\text{MIN}} \times 1.29\,\frac{\text{CU.IN.}}{\text{REV.}} \times 2 \times 0.95$$

WHERE:

1700 RPM = ASSUMED ELECTRIC MOTOR SPEED, FULL LOAD

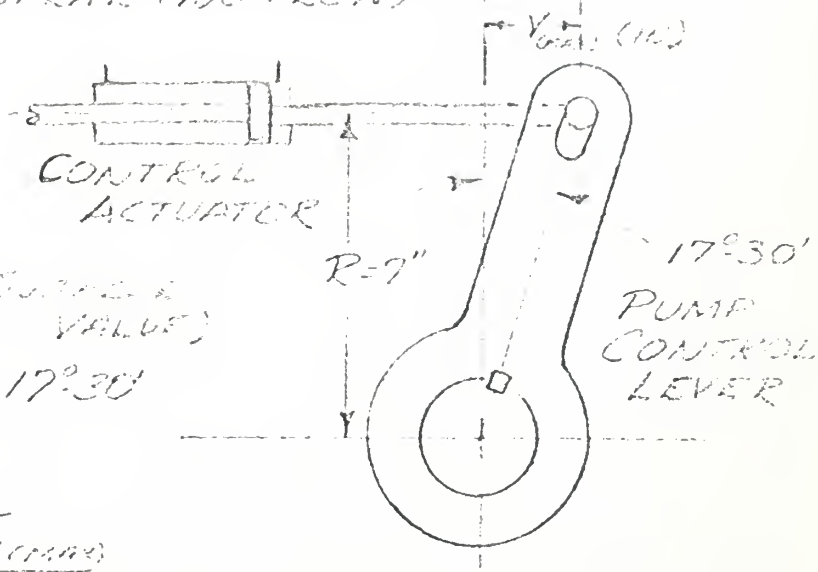
$1.29\,\frac{\text{CU.IN.}}{\text{REV}}$ = VOLUMETRIC DISPLACEMENT, MAXIMUM, EACH PUMP

2 = QUANTITY OF PUMPS

0.95 = ASSUMED VOLUMETRIC EFFICIENCY

$$F_{P(MAX)} = 69.5\ \text{CIS}$$

THE VARIABLE DISP. PUMP STROKE CONTROLS OPERATE THROUGH AN ARC OF 35°: 17°-30' EACH SIDE OF NEUTRAL (NO FLOW)



LET R = 7" (SUITABLE VALUE)

$$V_{(MAX)} = R \tan 17°30'$$

$$K_6 = \frac{F_{P(MAX)}}{V_{(MAX)}}$$

$$= \frac{69.5}{7(\tan 17\text{-}30')}$$

$$K_6 = 31.5\ \text{CIS/IN}$$

ASSUME PUMP FLOW VARIES DIRECTLY WITH CONTROL ACTUATOR STROKE

$$G_L = \frac{K_3 K_4 G_5 K_6}{1 + K_4 G_5}$$

$$G_L = \frac{(1.03 \times 10^{-4})(121.2)\left(\frac{2.55}{s}\right)(31.5)}{1 + (121.2)\left(\frac{2.55}{s}\right)}$$

$$G_L \approx \frac{1}{s + 309}$$

PARAMETER VALUES FOR SOLVING $G_{①}$:

$$L = \frac{W_1}{g} = \frac{648}{386} \approx 2 \ \frac{LBS \cdot SEC^2}{IN}$$

WHERE W_1 = 648 LBS EST WGT. OF ROTATING ASSEMBLY

$g = 386$ IN/SEC^2

A_m = "AREA OF PISTON", HYDR MOTOR

$$= \frac{\text{VOL. DISPL}}{\text{LINEAR DISPL.}}$$

$$= \frac{(25.4 \text{ CU IN})(.95 \text{ VOL. EFFICIENCY})}{2\pi(9.8 \text{ IN. WIRE LAYER RADIUS})}$$

$$A_m = 0.3925 \text{ SQ IN.}$$

$$M_p = \text{SOLID MASS} = \frac{W_2}{g} = \frac{664}{386}$$

WHERE W_2 = 664 LBS EST WGT.

$$M_p = 1.72 \ \frac{LBS \cdot SEC^2}{IN.}$$

$$M_F = \text{LIQUID MASS} = \frac{W_F}{g}$$

WHERE W_F = (68.76 CU IN ENTRAINED)(.0304 LB/CU IN)

$$M_F = \frac{(68.76)(.0304)}{386} = 0.0542 \ \frac{LBS \cdot SEC^2}{IN.}$$

$R = 1$ (ASSUMED)

$l = 60$ IN. (EST.)

$A = 1$ SQ. IN.

$\rho = 0.0304$ LBS./CU. IN.

$\beta = 4.16 \times 10^{-6}$ PSI^{-1}

REF: DESIGN OF HYDRAULIC CONTROL SYSTEMS, LEWIS & STERN, PG. 37

$\Delta Y = 60$ IN.

$E = 2.9 \times 10^{6}$ PSI

$D_o = 1.25$ IN.

$D_i = 1.125$ IN.

SUBSTITUTE ABOVE VALUES & SOLVE:

$$G_D =$$

$$\frac{(.0178)(1+2.82\times10^{8}s+3.20\times10^{15}s^{2}+4.65\times10^{21}s^{3}+2.40\times10^{29}s^{4}+1.56\times10^{36}s^{5}+1.04\times10^{34}s^{6})}{1+4.31\times10^{9}s+2.38\times10^{14}s^{2}+1.17\times10^{21}s^{3}+2.95\times10^{29}s^{4}+1.28\times10^{35}s^{5}+2.52\times10^{34}s^{6}+4.95\times10^{[illegible]}s^{7}}$$

$$G_{\substack{OPEN\\LOOP}} = \frac{K_1 G_D}{s}$$